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Prefabrication of Welded Piping for the Buzzard Point Plant

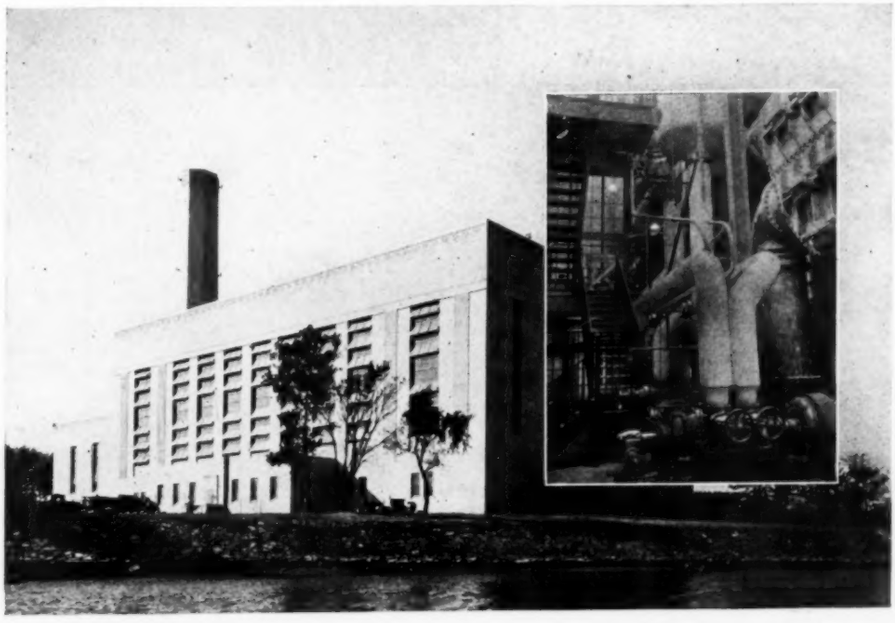
THE Buzzard Point station of the Potomac Electric Power Company—recently completed and put into operation to supply the increased power and light load imposed by the growth of our national capital with its many new buildings—is designed to operate at 730 lb steam pressure and a total temperature of 835 F at the boiler drums, producing 35,000 kw at present with basic design for expansion to 250,000 kw capacity. Into the design and construction of this station were incorporated many recent developments and refinements which it is expected will result in setting a new high mark for efficiency in a plant of this type consistent with sound engineering and long life. The piping is among these several features which show novel types of design and construction.

The high pressure steam piping is of flanged construction of the so-called Sargol type, seal welded for tightness. The pressure and temperature conditions called for high standards in this flanged construction.

[*Continued on page 244*]

• • • • *By* **A. W. Moulder***

***REFINEMENT IN THE ART** of welded piping calls for consideration of dependability (how to insure the quality of workmanship); smooth contours (with minimized friction losses); clean lines (absence of drip metal or "icicles"). How these objectives were attained in the installation of the piping at Buzzard Point is explained here....Because this piping job incorporates many novel types of design and construction, it serves as a particularly interesting example to show up-to-date practice in the welding of piping for severe pressure and temperature conditions.*



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The Buzzard Point plant of the Potomac Electric Power Company at Washington, D. C., recently put into operation to supply the increased power and light load imposed by the growth of our national capital, with its many new buildings. Stone & Webster Engineering Corporation were the builders. The inset is a more or less general view; the two insulated lines in the foreground are suction lines to the booster pumps from the 11th and 14th stage heaters

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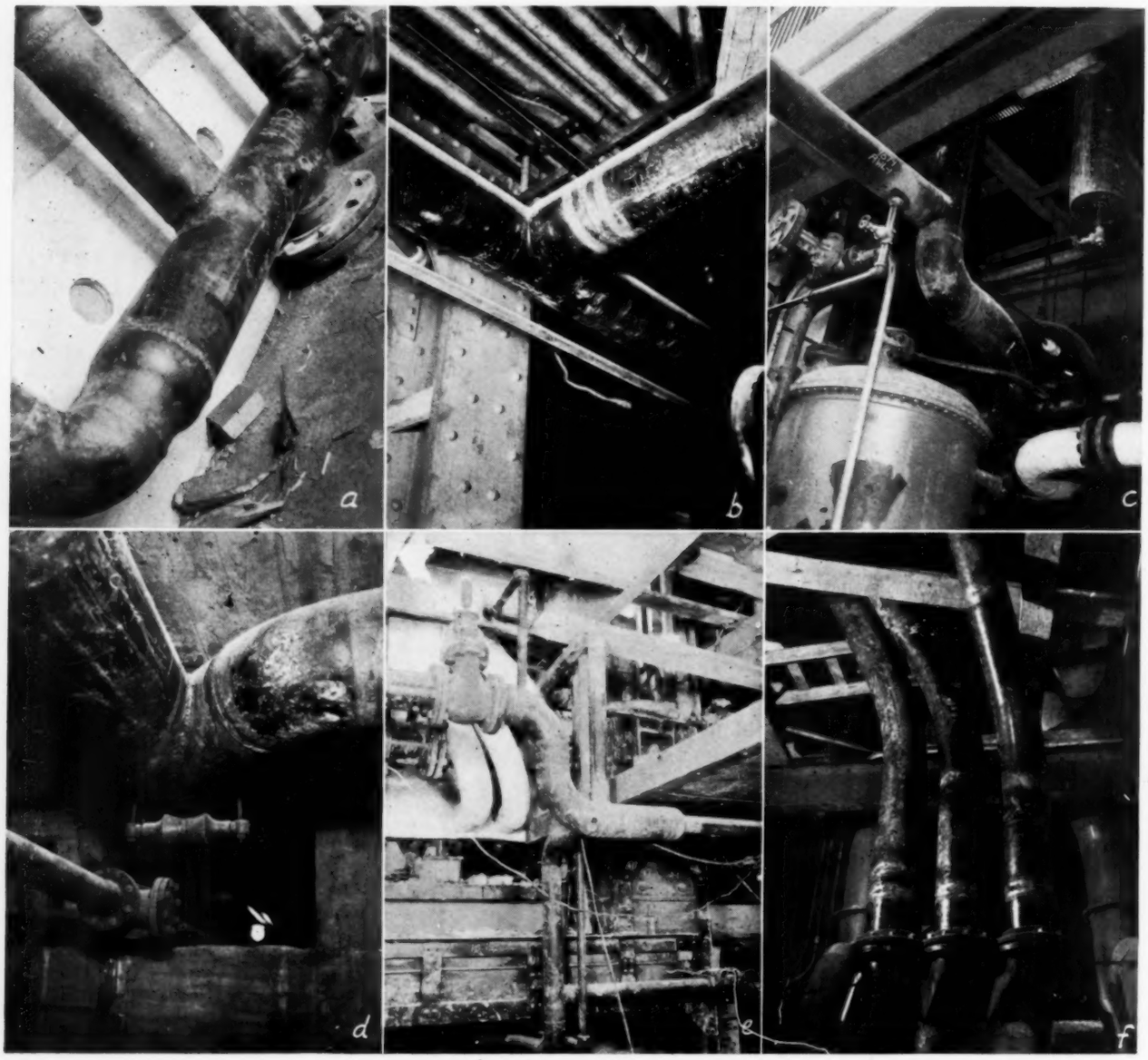
POTOMAC ELECTRIC POWER COMPANY'S BUZZARD POINT PLANT



a Cooling water to generator air coolers is carried in this piping.

The varied construction of this header makes it interesting. Two tee outlets on the top and one tee outlet on the bottom have been extruded from this single pipe. In addition forged steel threaded outlets are welded to the side for small screwed connections.

By noting the weld just visible in the upper corner of the picture it will be seen that this header was prefabricated with long outlet branches for cutting off to exact dimensions and making circumferential butt welds in the field.

b In this example a welding tee outlet of smooth contour and easy radius is extruded from the main pipe and a short outlet piece is welded to it. The exact length of outlet was unknown so a piece of approximate length was prefabricated to the extruded outlet allowing for cutting off to exact dimension in the field. This type of prefabrication made for straight, true workmanship on this project.

c In this picture is shown the blow-off tank with its connections, river water lines, boilerfeed pump suction and discharge lines, continuous blow-down piping, etc.

All-welded piping produced by prefabrication required a minimum of field welding.

d This drain from blow-off tanks located under the floor shows a type of welded construction requiring skill and proper equipment for alignment. Prefabrication and testing of such pieces reduced the field work to the more simple circumferential butt welds.

e This is water piping to slag gate at the rear of one boiler. It was surprising to find how sections of this kind could be advantageously prefabricated, the only limitations being the extent to which design is carried and the limitation of shipping and handling.

f Even the pulverized coal piping was prefabricated and welded into place. Pipe bends were designed to approximate dimensions with long ends and were cut off and welded into exact position.

INCORPORATES MANY REFINEMENTS—PIPING SYSTEM NOTEWORTHY

g Vent piping from oil pump, 11th stage heater, etc., illustrates the manner in which prefabricated welded piping can be produced to meet a variety of conditions. Note the forged steel threaded outlets and the extruded tee outlet to which is welded a drip pocket.

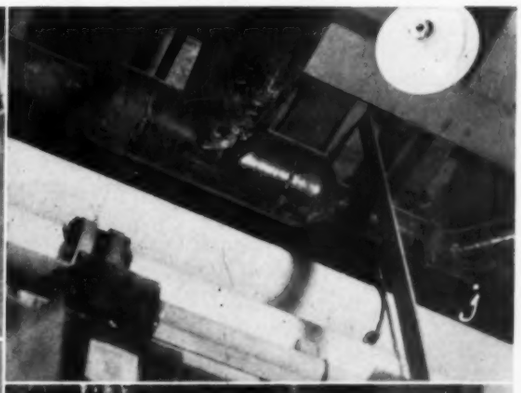
h The drain and over-flow from the blow-off tank are here joined together in a manner which is compact and economical in design but required skillful welding. Sections of this kind prefabricated by skilled welders with adequate equipment make for dependability. The flanges partly shown at the left are in the 1100-lb boiler feed water lines and are of the type with welded lap joints to be described in a later article.

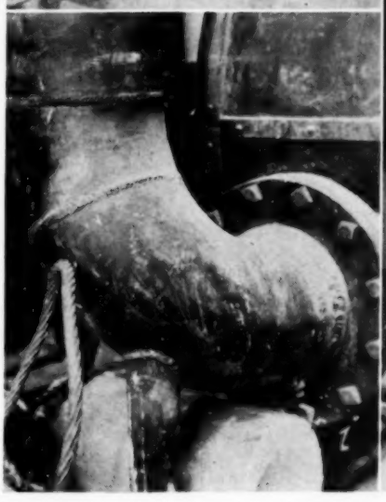
i This picture is the exhaust steam piping from turbine-driven boilerfeed pump. It illustrates how welding ells may be used to accomplish difficult and close offsets.

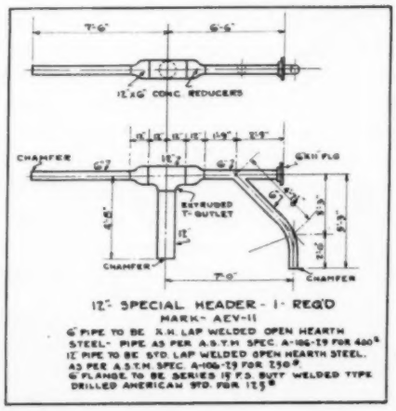
j The pop safety valve vent lines from evaporator are here accumulated into one short header piece. An excellent example of a smooth contour welded section, using a welding tee and concentric welding reducers all prefabricated into a section even greater than shown in the photo. An accompanying sketch shows prefabrication details.

k This header illustrates how prefabricated welded header construction may be used for connecting of several small screwed piping connections. Forged steel threaded outlets are here welded to a single header providing a good distribution manifold. By reviewing the pictures it will be observed that these forged steel threaded outlets, especially formed to fit the main pipe, are shown to be used to advantage in many places.

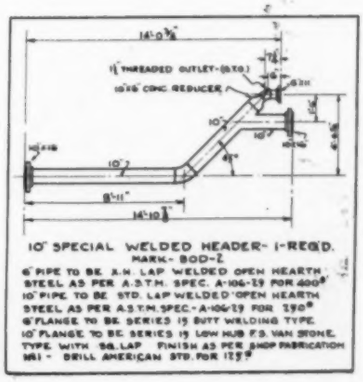
l Exhaust connections from boiler pop safety valves are tied into a header prefabricated with reduction in run and long outlet pieces.






12"x6" CONC. REDUCERS
CHAMFER
6"x11" FLG
EXTRUDED T- OUTLET
CHAMFER
CHAMFER
12"- SPECIAL HEADER - 1- REQ'D
MARK- AEV-11
6" PIPE TO BE A.N. LAP WELDED OPEN HEARTH STEEL- PIPE AS PER A.S.T.M. SPEC. A-106-29 FOR 400°
12" PIPE TO BE STD. LAP WELDED OPEN HEARTH STEEL. AS PER A.S.T.M. SPEC. A-106-29 FOR 250°.
6" FLANGE TO BE SERIES 15 F.D. BUTT WELDED TYPE DRILLED AMERICAN STD. FOR 125#

Shop sketch for prefabrication of section of piping, partially illustrated in photo "j"


10"x6" CONC REDUCER
10" SPECIAL WELDED HEADER- 1-REQ'D.
MARK- BOD-2
6" PIPE TO BE A.N. LAP WELDED OPEN HEARTH STEEL AS PER A.S.T.M. SPEC. A-106-29 FOR 400°
10" PIPE TO BE STD. LAP WELDED OPEN HEARTH STEEL AS PER A.S.T.M. SPEC.- A-106-29 FOR 250°
6" FLANGE TO BE SERIES 15 BUTT WELDING TYPE
10" FLANGE TO BE SERIES 15 LOW HUB F.D. VAN STONE TYPE WITH 5/8 LAP FINISH AS PER SHOP FABRICATION NO.1 - DRILL AMERICAN STD. FOR 125#

Shop sketch for prefabrication of section partially shown in photo "h"

Special alloy flanges were used, the alloy steel studs were tightened to definite elongation by micrometer measurement, and special flange covers were used. A refinement of practice such as to require the development of much new and valuable data resulting from careful engineering calculations and laboratory experimental work was included in this phase of the piping design, and will be covered in detail in a later article.

The piping other than for the higher pressures is of the more conventional type of welded construction. Even in this, however, new standards of design and construction have been followed and it is the purpose of this article to point out—pictorially more than by word—these features.

The value of welded piping construction from the standpoint of freedom from leakage, continuity of insulation, and in many cases installation cost, is well known. Refinement in the art of welding piping, however calls for carefully looking after the following:—

(a) Dependability—or how to insure the quality of workmanship.
(b) Smooth contours—with minimized friction losses.
(c) Clean lines—which means in welded piping absence of drip metal or "icicles."

The answer as formulated at Buzzard Point lies in the types of design and construction herein described.

Reasons for Prefabrication

Insofar as practicable all piping was prefabricated into sections before shipment so that circumferential butt welds only were necessary in the field. Even though the field welders were carefully picked and qualified by the strictest tests, it was deemed to be unfair to expect that even the best of these operators could compete under the handicaps of field welding with the quality of workmanship or finished product which could be produced in a plant equipped especially for this class of work.

Prefabrication permitted obtaining the advantages of the points mentioned above. First, dependability:—the prefabricated sections can be carefully tested before shipment. In addition difficult welding can be done under ideal conditions and with better facilities.

Prefabrication permitted minimizing the likelihood of drip metal or "icicles" in two ways:—First, by having most of the forming and cutting done under conditions where special machinery and equipment for such work was available instead of becoming necessary in the field where the varied skill of the welders is too often the controlling factors. Secondly, by careful shop inspection and cleaning where necessary.

Refinement in Design

Refinement in design is, of course, the direct answer to one of our items:—"smooth contour," to minimize friction. Of necessity refinement in design requires that the best available types of welding fittings (such as welding ells, tees, reducers—both concentric and eccentric—etc.) be incorporated into the design in the best and most economical manner.

Design is directly responsible for many phases of the problem, but must be coordinated with the possibilities presented by prefabrication to produce the desired refinements.

It is hoped that through the illustrations (pp. 242-243) and their captions, the reader may more fully realize the refinements possible by a proper coordination of design and prefabrication. It will be noted that the photographs were taken during the course of construction and in most cases the piping is shown before insulation was applied.

World's Fair Again Demonstrates

Popularity of Air Conditioning

AIR conditioned theaters and exhibits, air conditioned trains, air conditioned model homes, air conditioned restaurants, and the general discomfort inherent to any exposition on a warm day are again stressing the advantages of cooling for comfort to crowds attending the World's Fair, which re-opened in Chicago May 26. The 1934 Century of Progress is a bigger and better show in many respects, with new exhibits—the Ford Building illustrated on the cover is the outstanding one—and many improvements in last year's attractions.

Most of the air conditioning installations described in articles in H. P. & A. C. last year are still in service or on exhibit. In addition are air conditioned restaurants in the Black Forest Village, an air conditioned theater in the Ford Building, the Union Pacific's stream-lined air conditioned train, the C. B. & Q.'s "Zephyr," and others.

For a forceful demonstration of air conditioning's popularity, a little eavesdropping in a conditioned area at the Fair is hard to beat.

A view of the new Black Forest Village at a Century of Progress. Features include air conditioned restaurants and an ice skating rink

Recent Developments in SOUND CONTROL *and their importance in* AIR CONDITIONING

By **Vern O. Knudsen***

ABOUT TWO YEARS AGO Professor Knudsen wrote an article for HEATING, PIPING AND AIR CONDITIONING on how sound is controlled which has been widely quoted, referred to, and reprinted. Because there have been a number of developments since then in noise measurement and reduction which affect the air conditioning engineer, we asked Professor Knudsen to sum them up, bring us up-to-date. The present article is the result.... As the author points out, noise reduction concerns us in several ways. Noise produced by heating and air conditioning equipment must be prevented or reduced; as noises from other sources are reduced this will become even more important. Also, temperature and humidity affect the acoustics of rooms, and consequently an "acoustical asset" is added to the many others which come from air conditioning.

HUMAN efficiency and comfort suffer immeasurably from the noises incident to modern life, and consequently nation-wide efforts are being made to relieve the public from noise nuisances. The air conditioning engineer cannot escape, even if he would, the responsibility placed upon him in the campaign against noise.

Why Noise Reduction Concerns Air Conditioning

Noise reduction concerns the air conditioning engineer in two significant respects: (1) the noise produced by heating, ventilating, and air conditioning equipment is already, in many buildings, an irritating disturbance, and (2) as noises from other sources are reduced, as they surely will be, the noise from such equipment will become more conspicuous, and consequently this noise will become a target for protests. The engineer must therefore be alert to every development in noise reducing devices and materials which will help to secure quiet in buildings.

Besides interfering with speech and all other useful sounds, noise wears severely upon the nervous system, resulting in lowered mental productivity and probably even in a shortened tenure of life. It perhaps will be difficult to ascertain just what injury and loss of efficiency noise imposes upon us; undoubtedly certain individuals are affected by it more than others, but there is abundant evidence that all of us bear the injuries of this annoyance.

When we contemplate how successfully the noise of the gas engine has been eliminated in the modern automobile we have good reason to anticipate the elimination of other disturbing noises. And the ultimate reduction of automobile noises has by no means been achieved. Acoustical engineers, working with automobile manufacturers, are making continued efforts to further reduce the noise of the automobile. Only a few months ago a paper was presented before the *Acoustical Society of America* which described special acoustical filters which reduce the exhaust and intake noises of the automobile motor by as much as 15 to 20 decibels.[1]

Standardization of Noise Units

One of the prime needs in the scientific and practical approach to the noise problem is the standardization of units, scales, and instruments for the measurement of noise. Within the past two years there has been notable development in methods of rating and measuring noise. In January, 1932, there was organized through the auspices of the *American Standards Association,* with the *Acoustical Society of America* as sponsor, a Sectional Committee on Acoustical Measurements and Terminol-

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[1] C. E. Nelson, "Exhaust Mufflers for Automobile Engines," *Jour. Acous. Soc. of America*, 5, 224 (1934).

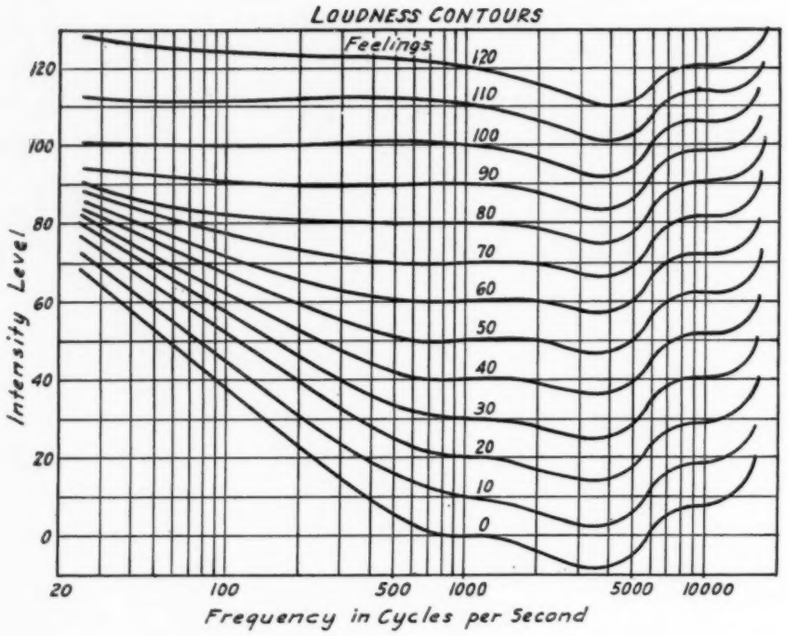

Fig. 1—Curves of equal loudness. Each curve has the loudness level indicated by the numeral above each curve. The β referred to in section 8 is given by the ordinates of these curves. (Fletcher and Munson.)

ogy. Already this Committee has made progress in standardizing units and scales for noise measurement, and a subcommittee is making progress on the standardization of noise meters. In the past, noise measurements made with one meter were not, as a rule, comparable with those made with another meter, and in some instances the same meter would rate one noise as greater than another, although the auditory judgment of listeners would rate the *lesser* noise (according to the meter) as the louder one! At a meeting of the Sectional Committee held in Chicago in June, 1933, and attended by twenty members representing twelve technical societies or industrial groups, the tentative standards proposed by the Subcommittee on Noise Measurements, with slight modifications, were adopted. The principal provisions of the proposed standards can best be made clear by quoting the following paragraphs: (The proposed standards are being circulated widely both in this country and abroad. Discussion is solicited from interested persons).

"1. The reference intensity for intensity level comparisons shall be 10^{-16} watts per square centimeter. In a plane or spherical progressive sound wave in air, this intensity corresponds to a root mean square pressure p given by the formula

$$p = 0.000207 \sqrt{\frac{H}{76}} \sqrt{\frac{273}{T}}$$

where p is expressed in bars,[2] H is the height of the barometer in centimeters, and T is the absolute temperature. At a temperature of 20 C and a pressure of 76 cm of Hg, $p = 0.000204$ bars.

(Note: This is approximately the root mean square pressure of a plane sound wave in air, having a frequency of 1000 cycles per second, which is just barely audible in quiet surroundings to the average person with normal hearing acuity).

"2. The intensity level of a sound is the number of db[3] above the reference level.

"3. The pressure level of a sound is 20 times the logarithm to the base 10 of the ratio of the pressure p to the reference pressure p_o. The unit of pressure level is the db.

"4. The reference pressure, p_o, for sound pressure measurements is 0.0002 bars.

"5. A plane or spherical sound wave having only a single frequency of 1,000 cycles per second shall be used as the reference for loudness comparisons. . . .

"6. The loudness level of any sound shall be the intensity level of the equally loud reference tone at the position where the listener's head is to be placed.

"7. In observing the loudness of the reference sound, the observer shall face the source which should be small and listen with both ears at a position so that the distance from the source to a line joining the two ears is one meter. . . .

"8. The loudness level of a pure tone propagated as a plane or spherical wave in air, and having a frequency of f cycles per second and an intensity level β db shall be defined by the set of curves given in Fig. 1."

"9. Until more accurate data are available, the relation between loudness and loudness level shall be that given by the curves shown in Fig. 2."

(Note: Fig. 2, prepared by Fletcher, is useful for converting the results of physical measurements on noise into the sensation of loudness, and provides a means for calculating, for example, the fractional reduction in loudness which results from a known reduction in intensity level—a matter of prime importance in the reduction of noise in ventilating equipment and in all problems concerned with the abatement of noise. A quantitative scale for loudness comparisons is thus available. For example, a loudness level of 70 db corresponds to a loudness of 8000, and a loudness level of 59 db corresponds to a loudness one-half as great, that is, 4000. Thus a reduction of 11 db in the loudness level, in the case here cited, reduces the sensation of loudness to one-half.[4]

The proposed standards, as is revealed by the curves in Fig. 1, will make the intensity level of the 1000 cycle reference tone the same as its loudness level. This is also approximately true for all frequencies between 500 and 10,000 cycles, but at frequencies below 500 cycles a certain change in the intensity level of a tone will be sensed as a greater change in loudness than that which results from the same change in the intensity level for frequencies above 500 cycles. This is particularly significant since the lower frequencies are predominant in the noises produced by ventilating equipment.

The foregoing quotations, curves and comments reveal the nature of the caution and understanding re-

[2]A bar is a pressure of one dyne per square centimeter.
[3]db—Abbreviation for decibel.
[4]For a thorough discussion of loudness and its measurement see article by Fletcher and Munson, *Jour. Acous. Soc. of Amer.*, 5, 79 (Oct., 1933).

quired in noise measurements and calculations. Obviously, the noise meter does not consist simply of a microphone, amplifier and ammeter, although these elements are basic in most meters. A reliable noise meter must incorporate suitable electrical or mechanical devices which will simulate the physical characteristics of the human ear. It is apparent, therefore, that for the present at least, any systematic program of noise measurement and noise reduction should be based upon a thorough foundation of physical and physiological acoustics. However, distinguished engineers on the research staffs of such concerns as the Bell Telephone Laboratories, the General Electric Company, and the Westinghouse Electrical and Manufacturing Company are engaged on the problems of noise measurement, and their efforts promise an early development of noise meters which will make it relatively easy for many engineers to cope with the problems of noise reduction.

Reducing Noise Transmitted Through Ducts

In an article the author prepared for HEATING, PIPING AND AIR CONDITIONING about two years ago,[5] practical methods were described for reducing both solid-borne mechanical vibrations and air-borne noise (such as is transmitted through ducts). These methods are effectively demonstrated in the recently completed Radio City project in New York. Not only air conditioning machinery, but walls, floors, ceilings, and entire rooms are insulated from the solid structure of the building by

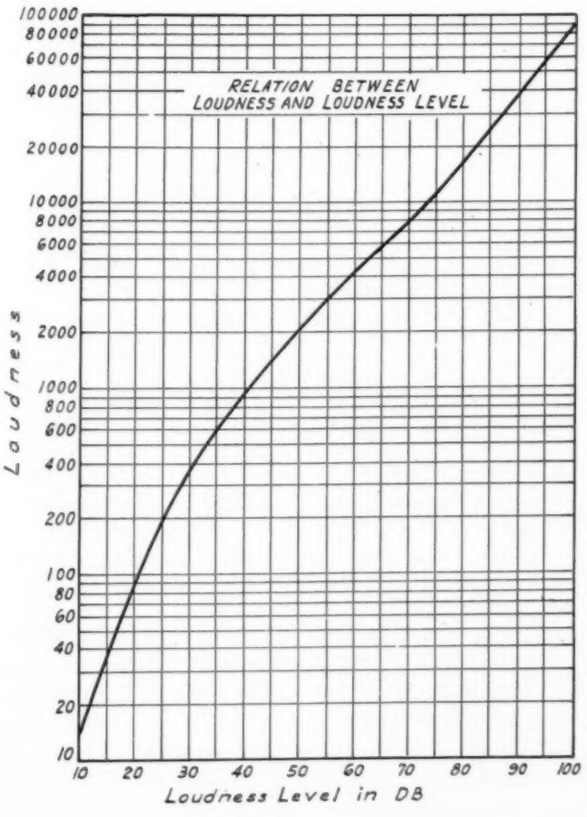

Fig. 2—Curves showing the relation between loudness (that is, loudness sensation) and loudness level. (Fletcher and Munson.)

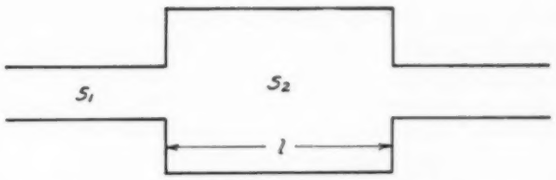

Fig. 3—Enlargement in a pipe or duct, which discriminates against the transmission of certain sound frequencies. (See *Eq. 1.*)

means of flexible steel clips, supports, and suspensions, *designed* with respect to elasticity and inertia so as to provide for all insulated machinery or rooms a low natural frequency compared with the frequencies of the mechanical vibration and noise which are to be insulated. Throughout the building, ducts lined with sound-absorptive material provide an effective insulation of the usual noises which are transmitted through ventilating ducts.

A striking demonstration of this method of absorbing sound is afforded visitors to Radio City. Two ducts, each about 1 ft square in cross-section and about 25 ft long, are set up side by side. One duct contains no lining material, and the other one is lined with a highly absorptive blanket. The guide speaks at one end of each duct while the visitors listen at the other end. When listening at the end of the unlined duct it sounds as if the guide were shouting directly in your ears, whereas you fail entirely to hear what he is saying when he is speaking through the lined duct. In fact, one suspects that he has lowered his voice until one goes to the other end of the duct and observes that he is talking equally loudly into each duct.

Changing Area of Pipe or Duct

Recently, two engineers of the Heinrich Hertz Institute, Berlin, devised a simple and an effective method for suppressing low frequency sound from being transmitted through pipes or ducts. The method is based upon the loss of energy in a sound wave which is reflected at a surface where there is change of area of cross-section in a pipe or duct. The theory of such reflection is fully described in Stewart and Lindsay's "Acoustics" (pp. 75-78). According to this theory the reduction in the intensity of a sound which is transmitted through such an enlargement[6] as is shown in Fig. 3, is

Decibel Reduction=

$$10 \log_{10}\left\{1+\left[\frac{1}{2}\left(m-\frac{1}{m}\right)\sin kl\right]^2\right\} \quad . \quad . \quad . \quad (1)$$

where m=the ratio of the cross-sectional areas S_2/S_1
l=length of the enlargement
$k=2\pi/L$, where L is the wave-length of the sound.

From this equation it will be seen that the attenuation or damping will be a maximum when $\sin kl = 1.0$, that is, when $l = \frac{1}{4} L (1 + 2n)$ where $n = 0, 1, 2, 3, \ldots$ In other words, if the enlarged portion of the pipe or duct be compared with a closed organ pipe, it

[5] "How Sound Is Controlled," by Vern O. Knudsen, HEATING, PIPING AND AIR CONDITIONING, October, 1931, p. 815.
[6] A constriction would give rise to a similar reduction.

will be noted that the attenuation is greatest for those sound frequencies which would be produced by a closed organ pipe having the same length as that of the enlarged portion of the pipe or duct. Thus, if the enlarged portion of the pipe be four feet long, the attenuation will be greatest for those sound frequencies having wavelengths equal to 16 ft, 16/3 ft, 16/5 ft, . . . By using two or more such enlargements in a pipe or duct, it is possible to discriminate against any required number of frequency components in a noise.

In rotating machinery, such as is used in the ventilating or air conditioning of buildings, the noise is usually made up of frequencies which are an integral multiple

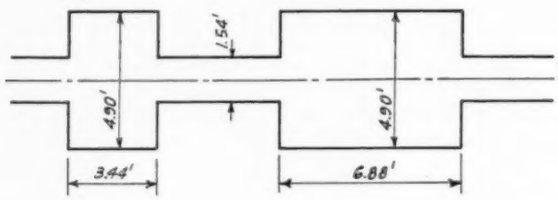

Fig. 4—Filter made up of two enlargements in a pipe or duct. See text. (Buchmann and Willms.)

of the speed of rotation, such as the fan-blade frequency, and a harmonic series of this fundamental frequency. Further, the fundamental frequency and the first four or five harmonics usually comprise almost all of the disturbing noise. Consequently, by suppressing these relatively few low frequency sounds it is possible to secure an entirely satisfactory elimination of the noise. For this reason, as few as two or three enlargements in a pipe or duct usually will be sufficient to provide a practical insulation for pipe- or duct-borne sound which otherwise would be disturbing. Thus, Buchmann and Willms[7] describe a two-enlargement filter (shown in Fig. 4) designed to suppress the noise of a certain machine, an analysis of which revealed that the principal frequencies consisted of 20.5, 41.0, 61.5, 82.0, and 102.5 vibrations per second. The enlargement in the area of cross-section is ten-fold, that is, $m = 10$. Any degree of enlargement can be used but from Eq. (1) it will be seen that m should be as large as possible. The insulation supplied by this device for frequencies up to 184.5 is shown in Fig. 5. It will be noted that for the range of frequencies which comprise most of the noise the attenuation amounts to approximately 20 db. This was found to be sufficient to furnish a practical elimination of the noise in this installation, and in general this amount of attenuation will correct most of the problems encountered in practice. The attenuation can be increased indefinitely by introducing more filters of the same type. For example, two sets of filters will provide approximately 40 db, three sets 60 db, and so on. Thus, when the frequencies to be eliminated are confined to a relatively narrow band of low frequencies, it is often feasible to suppress these frequencies by means of appropriately designed enlargements in the pipes or ducts.

[7]G. Buchmann and W. Willms, Schalldämpfung in Rohrleitungen (The Damping of Sound in Pipe-Lines), "Die Schalltechnik," March, 1933. (From the Heinrich-Hertz Institut für Schwingungsforschung).

Temperature and Humidity Affect Acoustics of Large Rooms

Within the past two years new data have been obtained on the effect of temperature and humidity on the absorption of sound in air, which have a bearing upon the problem of air conditioning and room acoustics. The curves in Fig. 6 show how the absorption of sound in air depends upon relative humidity at a temperature of 70 F. The absorption (attenuation) coefficient m, which depends upon temperature as well as humidity, and also upon frequency, enters into the formula for the calculation of the reverberation time in a room. The value of

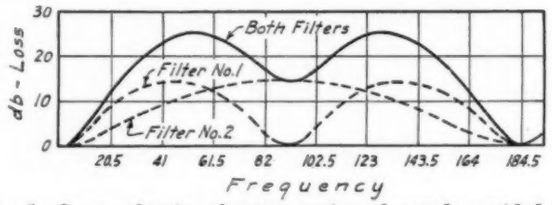

Fig. 5—Curves showing the attentuation of sound provided by the filter arrangement shown in Fig. 4. The attenuation shown by each filter separately is shown by the dotted curve, and the attenuation supplied by the two filters in combination is shown by the solid line curve. (Buchmann and Willms.)

m is so small at frequencies below 500 cycles as to be negligible in the calculation of reverberation for a room; but for frequencies above 500 cycles, especially at frequencies of 2000 to 10,000 cycles—frequencies which are extremely important for preserving the quality of speech and music—the reverberation time is markedly influenced by the absorption in the air. At a temperature of 100 F the coefficient m is approximately 40 per cent larger than it is at a temperature of 70 F.

The formula for the calculation of the reverberation time in a room is

$$t = \frac{0.049V}{-S \log_e (1-\alpha) + 4mV} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where

V = the volume of the room in cubic feet,
S = the interior surface of the room in square feet,
α = the average absorption coefficient of the interior surface of the room.

In large auditoriums, the reverberation of the high frequency components of speech and music is affected more by the condition of the air in the room than it is by the nature of the materials which form the boundaries of the room. Thus, at a frequency of 10,000 cycles, the value of m at 70 F and 18 per cent relative humidity is 0.020 per ft. If this value of m be substituted in Eq. (2) it will be seen that in the limiting case where the boundaries of the room are totally reflective, that is, where $\alpha = 0$, Eq. (2) reduces to $t = 0.049/4m = 0.62$ seconds. This would be the longest time of reverberation possible in a room at a frequency of 10,000 cycles when the temperature is 70 F and the humidity 18 per cent. But there is always some surface absorption in a room, including such absorptive objects as furnishings and audience, so that the actual reverberation time may be considerably shorter than 0.62 seconds. Of course, this low humidity is rarely, if ever, realized in a room occupied by people, but even at a relative humidity of 50 per cent with the temperature at 70 F, the longest

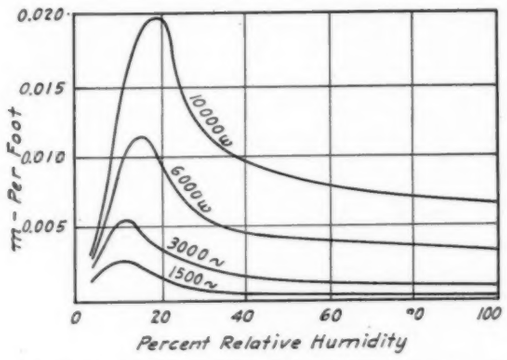

Fig. 6—Curves showing the attenuation constant per foot for sound waves traveling through air at different relative humidities, at a temperature of 70 F. The coefficient increases with temperature. Note that there is a particular humidity of the air which is more absorptive than any other humidity.

possible time of reverberation would be 1.30 seconds at 10,000 cycles; and, because of the surface absorption, probably would be not much more than two-thirds of this time.

It is obvious therefore that the air conditioning engineer should give consideration to the effects of humidity and temperature upon the outcome in the acoustics of large rooms. Fortunately, those temperatures and humidities which are regularly supplied by the proper air conditioning of a room are best adapted for providing good acoustical conditions. It should be a source of considerable satisfaction to the engineer to know that the proper air conditioning of a room is an essential factor in providing good acoustics in that room. For example, a large auditorium in a desert climate might suffer a serious acoustical defect by reason of the excessive absorption of sound in the hot and relatively dry air. Ideal acoustics in an auditorium is thus shown to require proper control of humidity and temperature, and therefore an acoustical asset is now added to the many other assets which come from air conditioning.

One Air Conditioning System LEADS TO ANOTHER

By **Henry W. Moore***

PROBABLY the first installation of a "refrigerated" air conditioning system in an apparel store in the Piedmont Section was installed last summer for the second floor of Ellis Stone and Company, ladies' ready-to-wear shop in Greensboro, N. C. It is of particular interest as it should mark the beginning of many more installations of cooling systems for high-class apparel stores in the eastern part of the South. The Southeast has been slower than other sections, the Southwest, for instance, in utilizing comfort cooling; heretofor the installations that have been made in the Piedmont Section, Virginia and the Carolinas, have been in the main theaters, restaurants, and five-and-ten-cent stores.

This Greensboro store has a 14-ton installation of the propeller fan, direct expansion, unit cooler type. One 7½- and one 5-hp compressors, using methyl chloride, are in use, together with 6 ceiling type unit coolers. The compressors are located in the basement, 150 ft from the farthest cooler, and are controlled automatically by room thermostats. The cooler fans operate continuously, independent of the compressors, to keep the air in circulation at all times.

The system was figured to maintain 85 F dry bulb and 40 per cent relative humidity inside, when the outside temperature is 95 F with 50 per cent relative humidity. This provides an effective temperature of 74.5 F inside, under these very hot outside conditions.

Advantages of Air Conditioning

Prior to the installation of this system the second (top) floor of this store was almost unbearably hot, and customers would not remain to make purchases. Now the floor, which is 150 ft long and 35 ft wide, is maintained 8 to 12 F cooler than the outside temperature during hot weather, and the store advertises it as "where spring is spending the summer."

Last July 1 this store purchased the system for their ready-to-wear department only, which occupies 2/3 of the floor. This department sells fur coats and dresses. The system was installed for this fashion floor primarily to increase summer sales of fur coats. After the system was put in operation the sales in this department increased immediately. Such favorable comments came from the customers that two weeks after the first job was completed the contract for cooling the rest of the second floor was awarded, which included the children's department and office.

The second installation was completed September 15 and the entire floor was satisfactorily cooled for the brief remainder of the summer. This is another example of how one successful cooling system leads to another, and how quickly stores will invest in air conditioning after the profitable results become apparent.

The owner of the store was well pleased with the power cost of operating this system, which amounted to approximately 28c per hr for electricity to operate the entire system. The increase in the store's power bill for the first month's operation amounted only to $35.00. However, the weather was not extremely hot in that month and it will doubtless run higher in warmer months.

A cooling tower was installed on the roof and the store advises that they found no appreciable increase in their water bill after the first month of operation.

*Manager, Greensboro (N. C.) office, Ilg Electric Ventilating Co.

New central heating plant in Washington, D. C. The stacks are housed in stainless steel enclosures so that the roof line presents an even and harmonious aspect

[*By* Pierce Timmis*]

New $3,800,000 Central Steam System

Heats 42 Washington Buildings

THE decision to bring together the many scattered government departments by housing them in new buildings in the "Triangle" in Washington, D. C., brought to a head the problem of providing steam for heating the old as well as the proposed buildings in this area.

As the new buildings were to be imposing and dignified with exteriors mainly of limestone, it was felt that everything possible should be done to aid in their preservation. It was thought that the presence of individual plants in this area with smokestacks ending at the roof line would, under certain atmospheric conditions, become a nuisance to the occupants of the buildings and hasten the blackening and decay of the stonework. Individual plants constructed at different times from different designs would have varying degrees of operating efficiency. Furthermore, the problem of trucking fuel and ashes for the individual plants through the streets was deemed to be an important consideration. It was believed that such unfavorable conditions could be minimized and economy could be secured through high operating efficiency and effective combustion control obtainable in a central heating plant located at the outskirts of the area. In a large plant of this kind devices which will eliminate practically all flying particles resulting from combustion may be effectively applied.

Following a complete study and analysis of loads, distribution and accessibility of fuel, the decision was made to construct such a plant to serve the new and old federal buildings in or near the Triangle and, later on, the federal and semi-federal buildings in West Potomac park.

The plant is located on the south side of C Street, S. W., between 12th and 13th Streets, adjacent to the 14th St. freight yard of the Pennsylvania Railroad as this is the nearest available site to the center of load meeting all requirements. The plant building is of brick and of a design consistent with that of other federal buildings in the vicinity. The stacks are not visible above the parapet line except at a considerable distance from the plant. They are housed in stainless steel enclosures so that the roof line presents an even and harmonious aspect from any point of view.

Maximum Steam Demand of 850,000 Lb per Hour

The maximum steam demand expected from the plant was estimated to be about 850,000 lb per hr when all buildings included in the ultimate scope of the project were added. To supply this demand six boilers, each having approximately 25,000 sq ft of heating surface designed for operation at 200 per cent of rating, were chosen, five for the expected ultimate demand plus one additional, which it was expected would be necessary as the development progressed.

The boilers are stoker fired and of the cross-drum inclined straight-tube sectional-header type, designed for a maximum working steam gage pressure of 400 lb per sq in. For the time being, however, they will be operated at 200 lb per sq in. Water walls are provided for the lower part of the sides and all of the rear walls of the furnaces. The side walls above the water-cooled sec-

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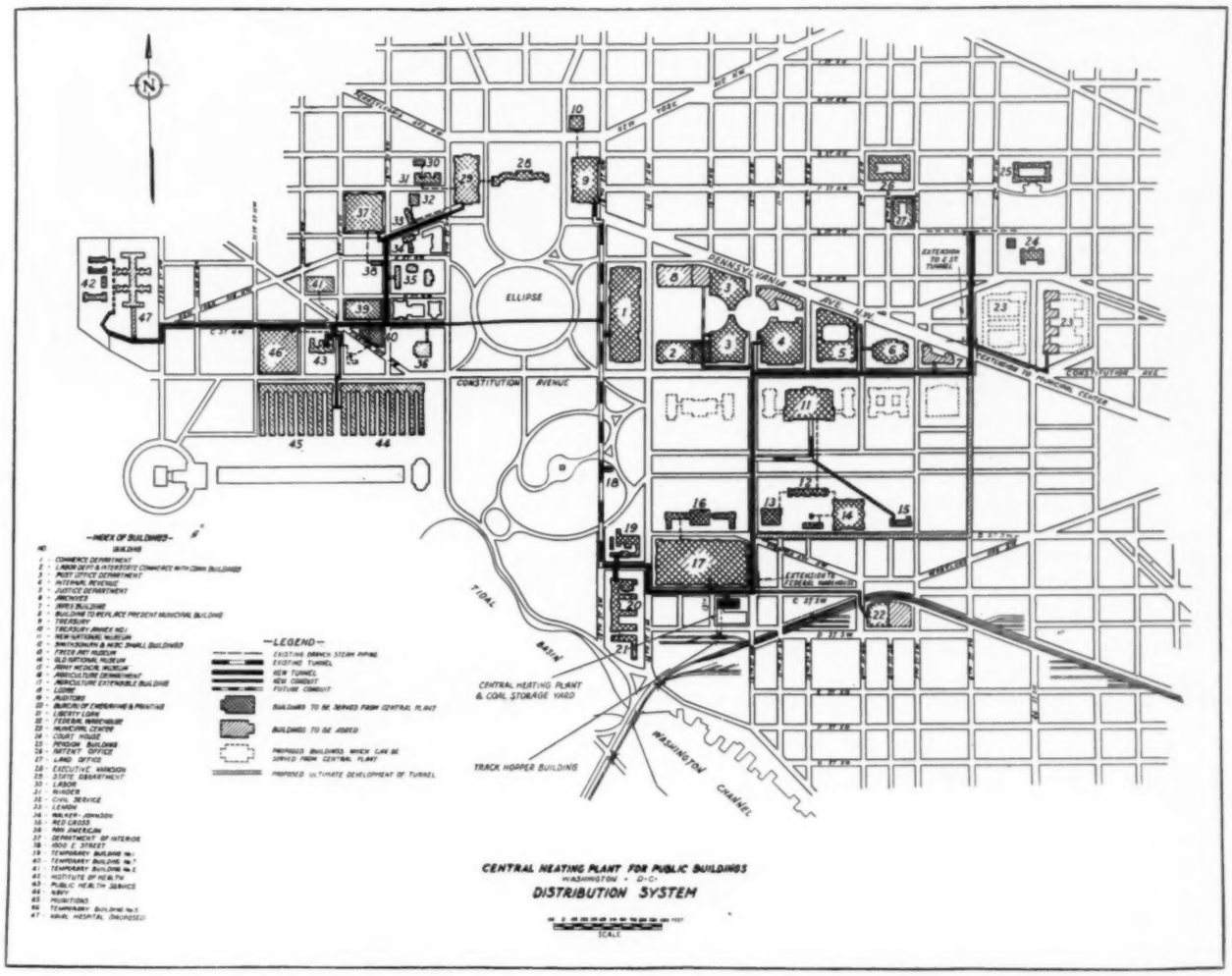

***FORTY-TWO BUILDINGS** varying in size from 25,000 to 27,000,000 cubic feet are heated by the new central plant described here. Cost of the project was $3,800,000—$2,200,000 for the plant and $1,600,000 for the distribution system . . . . . The author discusses the reasons for undertaking this project, describes the plant itself, the distribution system, and other interesting features.*

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tion are air-cooled and air for combustion is drawn down through them by the forced draft fans. Each boiler uptake is provided with a cinder trap to eliminate solid matter in the flue gases which pass through it. The induced draft fans discharge into the base of steel stacks at which point space is provided for the installation of gas scrubbers or precipitators if such equipment is later found necessary.

The Distribution System

The distribution system of the central heating plant now serves forty-two of the buildings in the Triangle and West Potomac park areas and others will be added as they are constructed. The buildings vary in size from about 25,000 to 27,000,000 cu ft, and the maximum steam demands from a few pounds to 85,000 lb per hr. While most of the buildings are of the office type, they vary considerably in construction from frame and stucco war-time temporary buildings to the monumental stone and brick structures exemplified by the new Commerce Building and the Agriculture Extensible Building.

With the exception of the conduit run across the Ellipse south of the White House, all main steam distribution piping is run in tunnels and the branches from the tunnels to individual buildings in conduit. There are about 18,000 linear feet of tunnels ranging in cross section from 4′ wide by 6′6″ high to 8′ wide by 7′ high and approximately 9,000 linear feet of conduit with external dimensions of 2′6″ high by 2′5″ wide to 3′0″ high by 5′0″ wide.

Prior to a final decision on the route of the tunnel system and the location of the tunnels and conduit with relation to existing structures, a complete topographical survey of the proposed route was made, as no accurate coordinate survey then existed. This survey showed the location of all large trees, manholes in all street utility lines, and, in addition, the elevations of all conduits, pipes and cables entering or leaving the manholes so that

it was possible to plot with some approach to accuracy the underground obstructions to be dealt with. The number of uncharted interferences encountered during construction were remarkably few.

In laying out the distribution system, use was made of certain portions of existing tunnels which were built during 1915 and which were the result of an earlier scheme for a central heating plant. Such portions of existing tunnels may be identified on the key plan. Wherever possible, tunnels are located in parked areas between the sidewalks and buildings or through the grounds to avoid disturbing street surfaces.

Besides the usual manhole inlets to the tunnels for material and inspection, emergency exits are provided at intervals of approximately 1,200 ft. These exits, which also serve as ventilators for the tunnel, are provided with counterbalanced grating covers which open part way when the latch is released. The latch may be easily released from the tunnel side but requires a key for operation from the outside. Emergency exit lights are placed at the entrance to emergency exits in the tunnel.

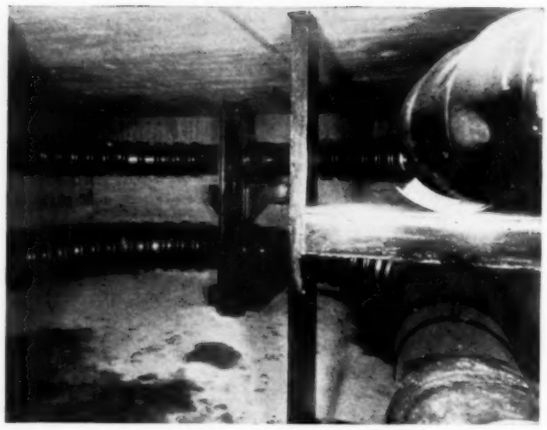
Sump pits receive drainage from the tunnels and are not connected with the tunnel proper except by a sealed drainage inlet. Each manhole housing expansion joints or traps, in the conduit system, is provided with two top inlets to aid natural ventilation or to permit the application of forced ventilation at times when work is being done in the manhole. The pumps are capable of unwatering tunnels in about 18 hours should they ever become flooded.

The distribution system is designed to carry steam at 300 lb per sq in. pressure, saturated, but 200 lb will be used for the present. There are three 18-in. steam lines leaving the central plant with provision for others. The total length of steam piping installed in tunnels and conduit and in the buildings is approximately 75,000 ft. In all buildings which existed at the time the system was installed, pressure-reducing-valve arrangements were provided to reduce steam pressure to the pressure formerly carried in the individual boiler plants. In the new buildings under construction pressure-reducing-valve arrangements were provided by others as part of the building contract. In the Treasury Building all floors but the fourth are heated with hot water, so hot water converters were installed to replace the coal-fired hot water boilers.

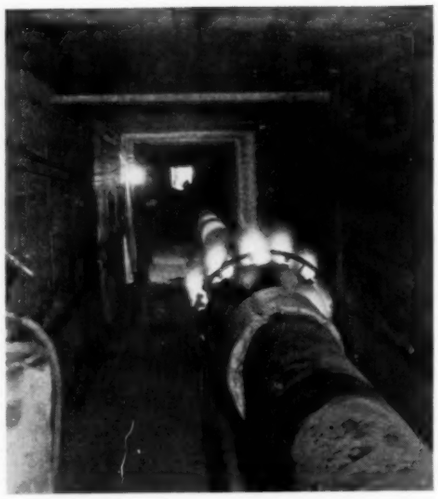
Condensate from all the Triangle buildings and all buildings south of the Triangle is returned to the central plant.

Piping Is Welded; Welds Normalized

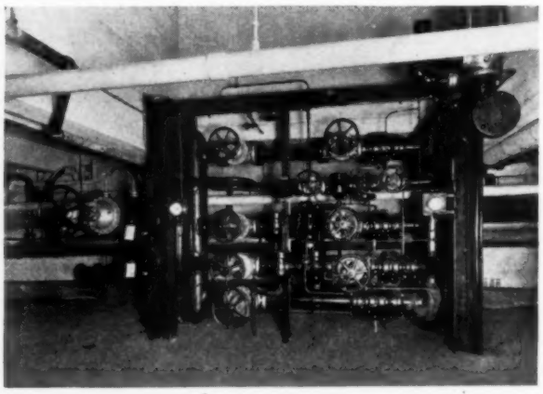
All pipe for steam and return lines in tunnels and for the steam lines in conduit is of seamless steel tubing 400-lb Standard. Condensate and trap discharge lines in conduits are of Grade "A" unfinished brass piping. All piping 1½-in. and larger is welded and all welds are normalized. Normalizing was carried out by bringing the temperature of the finished weld up to about 1,200 F and allowing it to cool slowly under a blanket of insulating material. A photograph shows one of the welds being heated by a gas furnace. Van Stone flanges are used at valves and expansion joints. All welds in piping 6 in. and larger in the tunnels are reinforced by sleeves



Expansion loops of corrugated pipe are used wherever possible. This view shows arrangement of the loops and the sliding supports



Normalizing a pipe weld to remove stresses by heating with gas furnace



Pressure reducing valve station before insulating, Agriculture Extensible Building

or straps, electrically welded to the pipe. The purpose of reinforcing the welded joints was to prevent a complete break in the remote possibility that a weld should fail. All welding was done to meet the highest standards for this class of work, by men qualified through welding tests.

Expansion loops of corrugated pipe were used wherever possible to provide for the expansion and contraction of piping. Where this was not possible slip tube externally- and internally-guided expansion joints were used except when the length of pipe involved was less than 60 ft, in which case packless expansion joints of the diaphragm type were used. Where elbows were required they were formed on the end of a pipe length by creasing, to reduce the number of fittings and the amount of welding.

Guides for piping are of the sleeve type in which a four-arm spider bolted to the steam line is free to move laterally with the pipe along a circular sleeve. The sleeves are spaced about 35 ft apart and are anchored to the tunnel construction. This arrangement insures good alignment between expansion loops or joints which, in general, are 400 ft apart.

All high pressure steam piping is insulated with three inches of felted asbestos put on in two 1½-in. layers, each layer being attached with copper binding wire. The outermost layer of felted asbestos is protected by an asphaltic impregnated asbestos felt. The binding rings of copper wire are spaced not more than 6 in. on centers. Low pressure piping and return lines are installed with two layers of insulation each 1 in. thick, also protected by asphaltic impregnated felt.

The drainage system for the piping consists of inverted-type steam traps located at anchor points. The inlet for the trap is taken from a drip pocket on the underside of the pipe or from the low point between the ends of the sleeves in a double-slip expansion joint. In some cases the traps discharge directly to the main condensate return line from the buildings and in other cases back to receiving tanks to buildings along the route.

Metering in Two Stages

Integrating steam flow meters are used for measuring steam to the buildings. Except in two or three cases metering is done in two stages; the base load up to 25 per cent of the maximum demand is taken on one meter and all loads above this point on another. This insures greater accuracy in the lower ranges, which is particularly desirable as the daily and seasonal demands vary greatly in most of the buildings.

The steam pressures at the three terminal points of the system, that is at the junction of 6th and E Streets, N. W., at the Treasury Building and at the Naval Hospital, are recorded on instruments in the central heating plant so that pressure drop may be observed.

The pressure reducing valves installed in the buildings under this contract are of the auxiliary pilot-operated diaphragm type designed to open against steam pressure. Diaphragms are of phosphor bronze. In most of the buildings two or three reducing valves are provided in parallel to take care of wide ranges in demand. Practically all reducing valve stations had to be installed in congested spaces so that the compact design illustrated in an accompanying view was used without sacrificing accuracy or steadiness of valve operation. In some cases where terminal steam pressures of 5 lb or less were required, as for instance, to the converters for the hot water heating system in the Treasury Building, pressure reduction was made in two stages.

Sump pumps in all sumps draining tunnels are in duplicate and are of the vertical centrifugal type. Wherever the pump motors and starting equipment are exposed to possible inundation, they are of special design to permit continuous operation when submerged.

Lighting the Tunnels

In general the tunnels are lighted by 50-watt clear lamps in vaporproof globes with guards, spaced not more than 100 ft apart. In particular they are spaced so that there is a light at all expansion joints, manholes and exits.

As the tunnels are of great length, the lighting supply is taken from several buildings along the route. Some of the circuits are over 2,000 ft in length so that four-wire, 3-phase circuits 120 volts, leg to neutral, are used to reduce voltage drop to a minimum. Separate circuits for providing power for the sump pumps are taken from buildings along the route in a similar manner.

The lighting circuits are controlled by contactors located in the several distribution centers and are operated by push buttons placed at convenient points in the tunnels and at all entrances and emergency exits. The push-button control switches for adjacent circuits are so arranged that the lights in the section ahead may be turned on and those in the section already traversed turned off.

The lighting was designed to provide enough illumination for general purposes. To provide lighting and portable tool service for maintenance of piping and apparatus, plug receptacles are placed generally at every other light, from which points any part of the tunnel may be reached with an extension cord 110 feet long.

Placing Radiator on Inside Wall

Is there any way to figure the difference in floor temperature with a radiator placed under a window and the same radiator placed on the inside wall of the same room with the same outside temperature? I know there is quite a difference in floor and ceiling temperature in a room of 10 ft ceiling with outside temperature at zero and the radiator on the inside wall.—W. A. N.

Answer: I do not know of any practical way to determine by computation the floor temperature difference between that with a radiator placed under a window and the same radiator placed on the inside wall of the same room with the same outside temperature.

It is well known that greater comfort generally ensues when the radiator is placed under the window than when it is placed against an inside wall opposite the window. The latter location, on the other hand, generally requires less heat input than does the under-window location, probably due to radiant heat entering the outside wall when the radiator is under the window.—Samuel R. Lewis.*

*Consulting Engineer, Chicago. Member of Board of Consulting and Contributing Editors.

New

Pressure Piping Code

Reviewed . . . .

by **Sabin Crocker***

IN continuing our "preview" of the Proposed American Standard Code for Pressure Piping, we consider this month the sections relating to fabrication details, and expansion and flexibility.

In January, the Code requirements for power piping were discussed, computing the thickness of pipe was considered in February, and the March and April installments were devoted to the sections on oil piping for the production, transmission, and refining of petroleum. The May article referred to gas and air piping, and district heating piping. The final installment of this series is contemplated to cover the requirements for welding.

Section on Fabrication Details

THE purpose of the Section on Fabrication Details is to provide a set of minimum requirement standards for: (1) the fabrication of hangers, supports, and the like; (2) the fabrication of piping joints other than welded; (3) the welding of piping joints; and (4) the provision for expansion and flexibility. Each of these divisions, which is termed a chapter, covers in considerable detail the minimum acceptable design, materials, and methods of fabrication for these items as applied in any section of the Code. Where a choice of method is available, it is left to the individual Code section to specify where each may be used.

Hangers, Supports, etc.

All materials used in the fabrication of hangers, supports, anchors, sway bracings, vibration dampeners, and the like, must be capable of meeting the test requirements of the designated *A. S. T. M.* specification for the respective material. The dimensions, physical properties, and chemical characteristics of these materials shall be such that, under the operating conditions for which they are intended to be used or to which they might accidentally be subjected, they shall be capable of sustaining such loading in accordance with the following factors of safety. With the exception of springs, all parts of equipment included in this chapter shall have a factor of safety of five (5) based on the ultimate strength of the material. Springs shall be fabricated so that the proper working fiber stress, depending upon the type of spring and the service for which it is intended, shall not be exceeded. If any part of the equipment is subjected to abnormally high temperatures, the resulting variation in ultimate strength of the materials shall be taken into consideration in the design.

All permanent hangers, supports and anchors shall be fabricated from durable materials suitable for the service conditions involved and where subject to atmospheric or other conditions likely to cause deterioration or corrosion shall be protected from such harmful conditions by painting or other suitable preservative treatment.

In tunnels and buildings of fireproof construction the use of wood and wire for piping supports shall be limited to rigging and temporary supports during erection and all permanent supports shall be of non-combustible materials.

In buildings of non-fireproof construction and in areas outside of buildings, piping may be supported by or hung from wood structures provided that all piping used for conveying fluids at a temperature above 230 F shall be so separated or insulated from such wooden members as to prevent dangerous heating.

Steel or wrought iron, except where other materials are expressly permitted, shall be used throughout in the fabrication of hanger rods, turnbuckles, beam clamps, pipe clamps and straps, chains, supports, rollers, guides, bases, and all other parts used for the support of piping.

Cast iron may be used for the fabrication of roller bearing bases, rollers, guides, anchor bases, brackets and parts of piping supports upon which the loading is mainly that of compression. Cast iron shall not be used for hanger rods, turnbuckles, clamps, or any parts which are subjected principally to tension loads. Malleable-iron castings of suitable design may be used for pipe clamps, beam clamps, hanger flanges, clips and bases, swivel rings, and similar parts of pipe supports, but their use shall be limited to cases where the operating temperature of the pipe line will not exceed 450 F. Nonferrous metals or corrosion resisting steel alloys should be used for supports under conditions causing excessive corrosion. Cast-iron or malleable-iron castings, if warranted as a protection against corrosion, also may be used subject to the limitations mentioned above.

Under conditions causing atmospheric rusting or slight corrosion, corrosion-resisting metals need not be used but a protective coating such as hot-dipped galvanizing, weather-resisting paint, or other suitable protection shall be applied after fabrication to all parts where required. Where no appreciable corrosion is apt to occur, corrosion-resisting materials need not be used but it is recommended that a suitable paint or other durable protective coating be applied to all parts after installation. Under any conditions, exposed screw threads on parts of this equipment where corrosion-resisting materials are not used shall be greased immediately after fabrication. Paints, slushes, or other suitable protective coatings may be used instead of grease.

*Engineer, The Detroit Edison Company, and Member of Board of Consulting and Contributing Editors. Among other standardization committee affiliations, Mr. Crocker is Chairman of Subcommittee 1 on Plan, Scope, and Editing of *A. S. A.* Sectional Committee *B 31* on Code for Pressure Piping.

This Month, the Sections on Fabrication Details and Expansion and Flexibility are Reviewed

Several paragraphs of interesting details are quoted verbatim:

"**Design Limitations. (a)** For spring hangers it is recommended that the springs be of the helical compression type, fitted top and bottom with bossed steel washers. Springs of types other than helical may be fabricated for use on spring hangers or supports provided their failure or mis-alignment will not completely release the piping load and that suitable provision is made to prevent excessive spring stresses.

"**(b)** The fabrication of helical and elliptical springs for spring hangers and supports shall be in accordance with the respective specifications designated. Springs shall be fabricated so that the proper working fiber stress, depending upon the type of spring and the service for which it is intended, shall not be exceeded.

"**(c)** All parts of the supporting equipment shall be fabricated and assembled so that they will not be disengaged by movement of the supported piping.

"**(d)** Pipe, straps, or bars of strength equal to the equivalent hanger rod may be fabricated for use instead of hanger rods for the support of piping of nominal pipe sizes 3½ in. and smaller.

"**(e)** Hangers used for the support of piping of 2 in. nominal pipe size and larger shall be fabricated to permit screw or equivalent adjustment after erection. Exception to this may be made in cases where hangers are to be used for the support of piping requiring exact grades, for which they may be fabricated as rigid hangers.

"**(f)** Screw adjustments shall be fabricated so that all threaded members will have a true and complete depth of thread. The turn-buckle or adjusting nut shall have its full length of thread in service while in use and the amount of adjustment shall be plainly visible at all times. All screw or equivalent adjustments shall be provided with suitable locking features."

"**Dimensional Limitations. (a)** STRAPS. Straps shall be limited to the minimum dimensions of ⅛ in. thickness by 1 in. width for use in locations protected from the weather. The minimum strap thickness for locations exposed to the weather shall be ¼ in. As exceptions to this, where straps are to be used for supporting pipe of 1 in. nominal size or smaller, the minimum strap dimensions shall be $\frac{1}{16}$ in. thickness by ¾ in. width in locations protected from the weather, and ⅛ in. thickness in locations exposed to the weather.

"**(b)** HANGER RODS. Hanger rods shall be limited to a minimum diameter of ⅜ in. for supporting pipe of 2 in. nominal size or smaller. For supporting pipe of 2½ in. nominal size and larger, rods used in the fabrication of hangers which are to be located in places protected from the weather, shall be limited to a minimum diameter of ½ in. Where rods are to be exposed to the weather or other corrosive elements they shall be fabricated from non-ferrous materials or given protective coatings according to the conditions. The minimum rod sizes for these corrosive conditions shall be as given above for the non-corrosive condition.

"**(c)** CHAIN. Chain when used for hangers shall be limited to round stock having a minimum diameter of $\frac{5}{16}$ in. or equivalent for supporting pipe of 2 in. nominal size or smaller. For supporting pipe of 2½ in. nominal size and larger, chain used in the fabrication of hangers which are to be located in places protected from the weather shall be limited to round stock having a minimum diameter of ⅜ in. or equivalent. Rods for chain hangers shall be fabricated in accordance with Par. (b) above. Where such composite chain hangers are to be exposed to the weather or other corrosive elements, castings and chain stock as well as the rods and all parts shall be fabricated, where applicable, from non-ferrous materials or given protective coatings according to conditions. The minimum sizes of chain stock for these corrosive conditions shall be as given above for the non-corrosive condition.

"**(d)** BOLTED PLATE CLAMPS. Bolted plate clamps used in connection with rod or chain hangers, shall have a minimum thickness of $\frac{5}{16}$ in. for weather-protected locations and ¼ in. for places where these parts are exposed to the weather. The bolts used for these clamps shall be of the same diameter as the hanger rod, or ⅜ in. diameter as minimum when clamps are used with chain."

"**Supports Permitting Pipe Movement.** When expansion or contraction of piping will be produced by reason of conveying fluids at temperatures above 150 F or by other causes, hanger and supporting equipment shall be fabricated and assembled to permit the free movement of this piping. This should be accomplished by the use of long rod hangers, spring hangers, chains, hangers or supports fitted with rollers, machined blocks, elliptical or circular rings of larger diameter than the pipe causing contact only at the bottom, trolley hangers or equivalent. Rollers shall be accurately made, properly guided and shall roll freely. Hanger rods shall be provided with welded or forged eyes, pivoting clamps, "T" heads with suitable sockets, or equivalent. In all cases allowance shall be made for rod clearance to permit swinging without setting up severe bending action in the rods. When pipe is covered by insulation, a suitable metal shield properly constructed and secured to the covered pipe should be used to protect the covering at points where the pipe is supported on movable rollers or in larger diameter rings."

"**Spring Hangers, Guides, Sway Bracings, and Vibration Dampeners. (a)** The fabrication and assembly of this equipment shall be such that spring failure or misalignment will not completely release the piping load. This equipment shall not unduly restrain the piping from expansion or contraction.

"**(b)** Where a long run of vertical piping is subject to considerable thermal expansion, the supporting equipment shall be fabricated and assembled so that this run of piping shall be either (1) supported on springs or other flexible supports in such a manner as to prevent excessive load on the bottom supports when expansion takes place; or else (2) the bottom support shall be capable of taking the entire load imposed by expansion, by failure of supports above, or during erection. This run of piping shall be adequately guided with equipment properly fabricated and assembled to withstand such reactions safely."

"**Counterweights.** Counterweights when used instead of spring hangers shall be provided with stops to limit travel. Weights shall be positively secured with bolts or placed in suitable box compartments. The fabrication of chains, cables, hanger and rocker arm details, or other devices used to apply the counterweight load to the piping, shall be subject to the requirements for materials in this section."

"**Anchors.** Anchors shall be fabricated and assembled in such a form as to secure adequately the desired points in runs of piping in relatively fixed positions. They shall permit the line to take up its expansion and contraction freely in both directions away from the anchored point, and shall be so ar-

ranged as to be structurally suitable for the particular location, line and loading conditions in question."

Expansion and Flexibility

THE Chapter on Expansion and Flexibility, which is short and requires little explanation, is quoted practically in its entirety.

"Expansion. The thermal expansion of piping for the more common piping materials shall be calculated from Table 1. For material not included therein and for more specific data on particular materials, reference shall be made to publications of the Bureau of Standards and to the Symposium on Effect of Temperature on the Properties of Metals (1931).[1]

Table 1—Thermal Expansion of Piping Materials[a]

Temp F	Steel and Wrought Iron	Cast Iron	Copper	Brass and Bronze
32	0	0	0	0
100	0.5	0.5	0.8	0.8
150	0.9	0.8	1.4	1.4
200	1.3	1.2	2.0	2.1
250	1.7	1.6	2.5	2.7
300	2.2	1.9	3.1	3.4
350	2.6	2.3	3.7	4.1
400	3.0	2.7	4.3	4.8
450	3.5	3.1	4.9	5.5
500	4.0	3.5	5.6	6.2
550	4.5			
600	5.0			
650	5.5			
700	6.0			
750	6.5			

[a]Total expansion in inches per 100 ft above 32 F.

"Flexibility. (a) Piping systems shall be designed to have sufficient flexibility to prevent the thermal expansion from causing unsafe stresses in the piping material, excessive bending moments at joints, or excessive thrusts on equipment or at anchorage points.

"(b) Flexibility shall be provided by changes of direction or by the use of bends, loops, offsets, or expansion joints of the slip joint or corrugated types. Flexibility may be increased by corrugating the pipe wall.

"(c) In calculating the flexibility of a piping system, the straight portions of the system must be included as contributing to the flexibility of the system in conjunction with bends or other changes of direction of the line.

"(d) Where pipe lines join or where branches occur, provision shall be made for the expansion of both lines.

"(e) As an additional expansion precaution and to facilitate assembly, the pipe may be cut short by an amount approximately one-half the calculated expansion and sprung into place. No reduction in the expansion to be cared for shall be made because of such cold springing, except where a section of the line between anchors lies in one plane, in which case the effective expansion between anchors may be taken as not less than two-thirds of that computed from Table 1.

"(f) In calculating the flexibility of piping, allowances must be made for the flattening effect present in curved pipes."

"Supports. (a) Supports for piping shall be designed so as to allow for the free expansion and contraction of the pipe without causing excessive strains in the pipe or on the supports or anchors.

"(b) Supports for piping shall permit side movement of the piping where it is caused by the normal expansion.

"(c) Anchors and guides shall be provided and located so as to confine and guide the expansion in direction so as to use properly the flexibility of the system."

[Two chapters of the Code are devoted to the fabrication of pipe joints, the divisions being assigned to welded pipe joints and to joints other than welded. The latter sets forth requirements for threaded joints and the more common types of flanged joints. The Code, however, permits the use of special joints such as rubber or flexible rings held tight by pressure and properly retained, sleeve or gland type, hub and spigot, screwed or flanged union connections, etc., provided suitable materials are used for the service conditions encountered as required by other sections of the Code. Since the requirements for making flanged and screwed joints merely set forth what is generally known as good practice there is no occasion to recount them here. The requirements for welding of pipe joints are more novel and will be made the subject of the next and last installment of this series which will appear in the July issue.—*Editor.*]

[1]Sponsored by Joint *A. S. M. E.-A. S. T. M.* Research Committee on Effect of Temperature on the Properties of Metals, Chicago, Ill., held in Chicago on June 23, 1931. (Published by *A. S. M. E.-A. S. T. M.*).

Air Conditioning Necessary in Any "Decently Constructed" Building

By **F. J. Wilkins***

ANYONE who has experienced the advantages of air conditioning either in the home, place of business, store, restaurant, etc., realizes that the time is inevitably coming when air conditioning will be considered as necessary a part of any decently constructed building as are the walls and roofs.

There has perhaps been some tendency with the earlier installations to operate at too low a temperature. There is a point beyond which, in my opinion, it is not advisable to reduce the temperature and this may perhaps be safely stated as about 10 to 12 F below that outside of the building. Otherwise there is a distinct shock felt on entering the building or on leaving the conditioned area to upper floors which may not have that advantage.

It is noticeable that customers like to shop in the summertime where the air is properly conditioned, but the advantage in the winter is perhaps just as great because the air at this season of the year is quite apt to be more impure than in the summertime—especially in smoky, fog-ridden areas. The effect on the vitality of the salespeople is noticeable; those, for instance, in our basements, first floor and mezzanine seem distinctly fresher at the end of a summer day than salespeople on other floors.

Our restaurants on the 13th floor are air conditioned and it is noticeable that some customers and employees who do not normally eat there during the year do so during the hot summer months.

*Building Superintendent, The J. L. Hudson Company, Detroit.

Protecting Fans from Corrosion

THE housings of fans which handle certain kinds of gases are naturally subject to corrosion under the right conditions. Zinc or other suitable metal coatings can be applied to such surfaces for protection by using a metal spray process.

Some ventilating fan housings were recently protected in this way with a 0.01-in. coating of zinc; in cases where the conditions are very severe this may be tripled. The metal surfaces were prepared by sand blasting, and the metal coating applied with a portable spray gun into which the coating metal was fed in the form of a wire where it is melted, atomized and sprayed very like paint from a spray gun.—Francis A. Westbrook.

Heating *and* Air Conditioning Buildings *with the* Reversed Refrigeration Cycle

Gas-Engine-Driven Compressor Utilized in Proposed Layout

By C. F. Mowrey*

***IT'S A POOR RULE** that doesn't work both ways. We lose in converting heat into power, but gain in converting power into usable heat. We cannot, of course, effect a combination of engines and refrigerating machines that will give us something for nothing, but we can—practically speaking—obtain some low-temperature heat for nothing by utilizing the reversed refrigeration cycle, as discovered by Lord Kelvin in 1852.*

This article describes how a gas-engine-driven compressor (or heat pump) can be used to heat buildings by the reversed refrigeration cycle to obtain high efficiencies. The proposed arrangements, while unique, are sensible and involve equipment now available.

UNQUESTIONABLY, the reversed refrigeration cycle (heat pump) is going to enjoy further development and use, and it is to be expected that the industries interested in the sale of gas or gas-consuming equipment could derive considerable benefit thereby. Consequently it is the purpose of this article to point out that the development of heating and air conditioning systems utilizing gas in connection with this cycle should be well worth-while. There are, of course, problems to be considered, including cost of equipment, but any system of producing heat that will reduce the gas consumption by at least fifty per cent should certainly deserve careful consideration.

The arrangement of equipment as proposed in this article includes the paralleling of condensers and evaporators, the air being warmed in increments as it is blown through one condenser after another. It is possible by this method to obtain a practical coefficient of performance of greater magnitude than the apparent theoretical coefficient of performance. For instance, if the temperature of air is to be raised from 0 F to 100 F, the theoretical coefficient of performance is 460÷100 or 4.6. However, if a number of compressors and coils are arranged so that each evaporator picks up heat from the zero air, but No. 1 condenser raises the temperature of the air to be warmed from 0 F to 100 F to, say, 10 F and No. 2 condenser raises the temperature from 10 F to 20 F and so on, up to 100 F, then the coefficient of performance would be based on 460 divided by the average between 10 F and 100 F or 55, with proper allowances made for the actual temperatures in the coils. Therefore, for certain conditions, such a method of application could result in a greater actual coefficient of performance than the apparent theoretical coefficient of performance. The possibility of such an application is quite generally overlooked.

Heating by Reversed Refrigeration Cycle

For the benefit of those who never have had occasion to delve into this particular branch of thermodynamics, and others who have forgotten, an explanation of the reversed refrigeration cycle follows: Absolute zero, equivalent to —460 F, is considered to be the lowest temperature that could ever be obtained. If an object feels cold, we are apt to consider it as containing no heat, but if we bear in mind that any object at a temperature above —460 F contains heat, we are in position to understand what can be gained by the use of the reversed refrigeration cycle in heating. The so-called reversed refrigeration cycle is of course no different from the refrigerating cycle employed for the purpose of maintaining the proper temperature in a refrigerator. The term is used simply to establish the fact that the primary purpose is to deliver heat rather than to remove it, although of necessity the cycle must always accomplish both results.

A simple refrigerating system consists of a compressor; a coil, called the evaporator, leading to the compressor; a coil, called a condenser, leading from the compressor, and a pressure reducing valve between the two coils. If operating as a refrigerating system, the evaporator coil is placed in the refrigerator and the condenser coil is placed in such manner that air or water will flow over it. When the compressor is in operation, it draws

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the refrigerating gas from the evaporator coil, maintaining a comparatively low pressure and correspondingly low temperature. As a result, the temperature of the air in the refrigerator is higher than that of the gas in the evaporator, and heat flows from the former to the latter. The gas is then compressed into the condenser coil. This compression raises the temperature of the gas to a higher level than that of the air or water flowing over the condenser coil; therefore, the heat that was absorbed in the refrigerator flows to this air or water and is carried away. Liquefaction of the gas in the condenser coil occurs as a result of the reduction in temperature at the higher pressure. This liquid then passes through the pressure reducing valve (called a float valve or an expansion valve) into the evaporator coil at a comparatively low pressure and correspondingly low temperature to complete the cycle.

Figure 1—Opposite

Operation of the proposed compressor and gas-engine plant utilizing reversed-refrigeration cycle for heating. The plant can be operated for cooling as indicated on the diagram. All air is recirculated, and "efficiency" is 151 per cent

Consider a simple application of this system from the heating angle. Assume that a refrigerating system is to be used to heat a small building to 70 F when the outdoor temperature is 30 F. The evaporator coil would be outdoors and the condenser coil indoors. We could assume that the gas in the evaporator coil would remain at about 20 F, thus absorbing heat from the 30 F surrounding air and that the condenser coil would give up heat to the air in the building at a temperature of about 100 F. We will assume further that the plant would be sufficiently large to maintain a 70 F temperature in the building.

Performance charts based on ordinary refrigeration practice indicate that for the given conditions the power required to drive the compressor would be 1.2 actual hp for every 12,000 Btu's picked up from the outdoor air and delivered to the building. The heat equivalent of 1.2 hp is, however, 3,050 Btu, which indicates that there would be delivered to the building slightly more than 2½ times as much heat as the heat equivalent of the power required to drive the compressor. This does not take into consideration the type or cost of the motive power used, but simply serves to point out what could be done with present-day equipment as far as efficiencies or rather, coefficients of performance, are concerned.

Use of Gas-Engine-Driven Compressor

This article concerns the application of combustible gas to the reversed refrigeration cycle. Gas engines[1] (not including gasoline engines) are comparatively unknown in many parts of the country, but due to the increased availability of cheaper natural and mixed gas and the improvements that have been made in the two-cycle engine within the last five years, their use should increase.

As might be inferred, the two-cycle engine is much simpler than the four-cycle engine, but up to recent years its efficiency was very much lower than that of the four-cycle. Careful tests made of the operation of the modern two-cycle engine indicate a thermal efficiency of a little better than 25 per cent at full load. The sum of 250,000 hp can be developed by engines of this type in operation today, the major portion being used for driving direct-connected compressors in the oil fields. There are also in use a number of this type of engine directly connected to refrigerating compressors.

Let us consider what percentage of the theoretical coefficient of performance we might expect from a gas-driven compressor to be used for heating a building. First, the engine would operate at full load and under ideal conditions, which should result in a thermal efficiency of about 25 per cent. Secondly, considering that actual coefficients of performance of around 70 per cent of theoretical have been obtained from refrigerating plants for years and also considering that the mechanical efficiency of this type of compressor would be high, we should be safe in assuming that the actual coefficient of performance would average 60 per cent of the theoretical coefficient of performance. If we multiply 25 per cent by 60 per cent, we obtain 15 per cent, which represents the net percentage of the theoretical coefficient of performance we would expect to obtain, based on the heat input to the engine.

Application to a Warm-Air System

Assume the application of this type of heating in conjunction with a warm-air heating system. The lower the register temperature, the greater will be the efficiency of our system, so we will assume 120 F register temperature. Experience has demonstrated that with forced circulation this temperature is high enough to result in thoroughly satisfactory conditions in the central part of New York State, where the average temperature for January and February is 24 F and there are about 6,800 degree days in a heating season.

Although the average winter temperature in this climate is 40 F, inasmuch as the efficiency of the heat pump is dependent upon the temperature, we must take into consideration the average temperature at which the total yearly quantity of heat is supplied. An analysis of the degree-day record shows this can be obtained by dividing the sum of the squares of the number of degree days per day by the total number of degree days per season, and then subtracting this quotient from 65. The result will be about 30 F, which means if we determine the efficiency of the heat pump system for an outdoor temperature of 30, we will have determined the expected average efficiency for a heat pump operating in a climate such as that prevailing in the central part of New York State.

The proposed layout is shown in Fig. 1. *EC* is the gas engine and compressor unit. Outdoor air at 30 F is blown through the evaporators E_1, E_2 and E_3 by a fan driven by the gas engine. The 20 F gas in the evaporators absorbs heat from the outdoor air. This gas is then

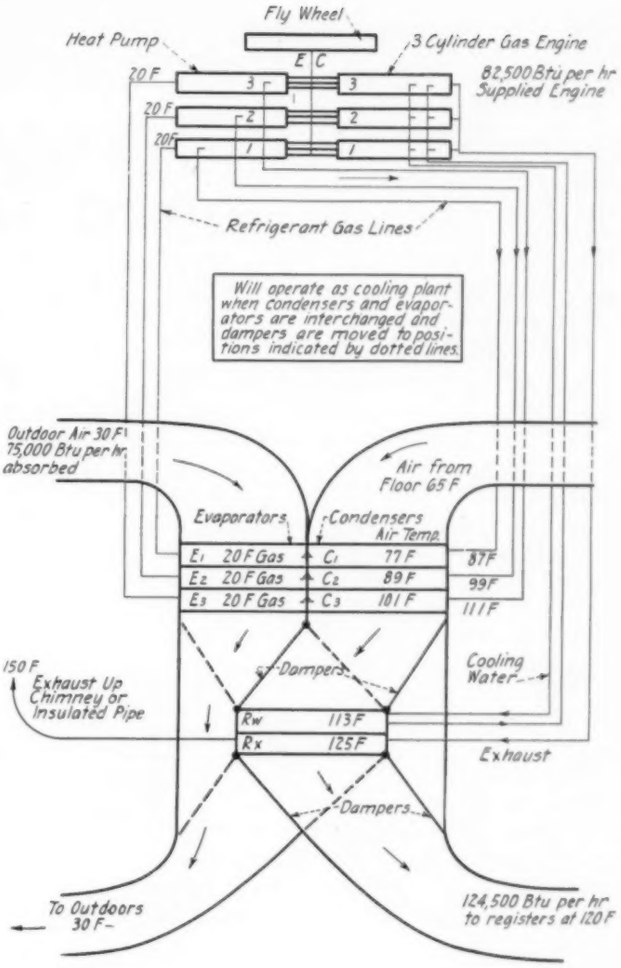
[1]Regarding the two-cycle fuel injection type of gas engine, particular attention is directed to the following publications: Pennsylvania State College Technical Bulletin No. 16, and "Natural Gas in the Refrigeration Plant," by F. W. Laverty, published in *A. S. R. E.* Convention Reprint, Jan. 26, 1932.

drawn through three separate pipe lines into three separate cylinders of the heat pump and compressed into the radiator condensers C_1, C_2 and C_3. The cooling water from the gas engine is circulated through radiator R_w, while the exhaust heat from the engine is circulated through radiator R_x. The air in the building is drawn from the floor at 65 F and blown through radiators C_1, C_2, C_3, R_w and R_x progressively and back through the registers at 120 F.

Before going into the question of control, let us see what would happen from a standpoint of heat balance for a certain assumed condition. We will assume the hourly building heat loss to be 100,000 Btu, when the indoor temperature is 70 F, and the outdoor temperature zero. To allow for pick-up, we will figure on a plant large enough to actually deliver 125,000 Btu per hr. As the outdoor temperature for this particular case is assumed to be 30 F, the plant will not run over 40 per cent of the time, but will deliver full capacity when it is operating.

We will estimate that for the given conditions 60 per cent of the heat is to be supplied by the heat pump; that is, at the rate of 75,000 Btu per hr. In order to determine the theoretical coefficient of performance, we will assume that the air delivered to the registers at 120 F leaves the last radiator at 125 and further that as the air is heated from 65 to 125, 60 per cent of the heat is supplied by the heat pump and the other 40 per cent by the heat in the cooling water and exhaust radiators R_w and R_x, respectively. In that event, the heat pump would have raised the temperature of the air 60 per cent of the way from 65 to 125 or to 101 F. The air would have been raised progressively as follows: by C_1 from 65 to 77; by C_2 from 77 to 89 and by C_3 from 89 to 101 F. If the temperature of the gas in each condenser is 10 F higher than the temperature to which it raises the passing air, the gas temperatures must be 87, 99 and 111, with an average of 99. Consequently, the coefficient of performance would be based on the temperatures of 99 and 20 and would be $\frac{460+20}{99-20}$ or 6.01. We previously calculated that the actual coefficient of performance would be 15 per cent of the theoretical coefficient of performance, based on the heat input into the engine. Therefore, the actual C.P is 0.15×6.01 or 0.91. If the 75,000 Btu delivered by the heat pump represents 91 per cent of the heat supplied the engine, the total heat supplied must be 82,500 Btu. On that basis we must obtain 50,000 Btu from the "waste heat" of the engine in order to bring the total to 125,000 Btu.

Consequently, the next question is the percentage of "waste heat" that may be recovered. Tests indicate that about 25 per cent would go into the cooling water, and about 38 per cent into exhaust heat. All of the heat going into the cooling water could be utilized. It should not be difficult to utilize two-thirds of the exhaust heat, which would be about 26 per cent of the heat input. The 25 per cent of the heat input that goes into power is ultimately converted back to heat and mostly at the heat pump. We will consider that 0.4 of this 25 per cent could also be utilized as heat, which means an additional 10 per cent. This makes a total of 25% + 26% + 10% or 61%—, say 60%. Sixty per cent of the 82,500 Btu



supplied equals 49,500 Btu. Consequently, there would be delivered to the building 124,500 Btu for the 82,500 Btu supplied the gas engine, which would result in an "efficiency" of 151 per cent, better than double that of the average gas-fired furnace.

Size of Engine and Compressor

The size of the heat pump required, expressed in tons of refrigeration, based on the standard of condensation at 86 F and evaporation at 5 F, would be about $\frac{75,000}{12,000}$ divided by 1.3, or 4.8—, say a 5-ton capacity pump. Inasmuch as the efficiency of the gas engine is taken as 25 per cent and 1 hp equals 2545 Btu, we can say that 1 hp would be produced for every 10,000 Btu supplied the gas engine, which indicates an 8.8-hp gas engine for this case.

The capacity of a refrigerating machine decreases as the temperature of the gas in the evaporator is lowered. Let us see if the compressor and engine would be large enough for zero weather. A cut-and-try method shows that for zero weather the heat pump would supply about 52 per cent of the heat, or 65,000 Btu per hour. At the reduced capacity the heat pump should have a capacity of $\frac{65,000}{12,000}$ divided by 0.65, or 8.33 tons. The "waste

heat" to be supplied by the engine would be equal to 125,000 Btu—65,000 or 60,000 Btu. Figuring that 60 per cent of the input can be utilized as waste heat, the input would be $\frac{60,000}{0.6}$ or 100,000 Btu. Therefore, the size of the gas engine would be $\frac{(100,000)}{(10,000)}$ or 10 hp. It is apparent that this building would require an 8.33-ton compressor driven by a 10-hp gas engine.

Possible Control Methods

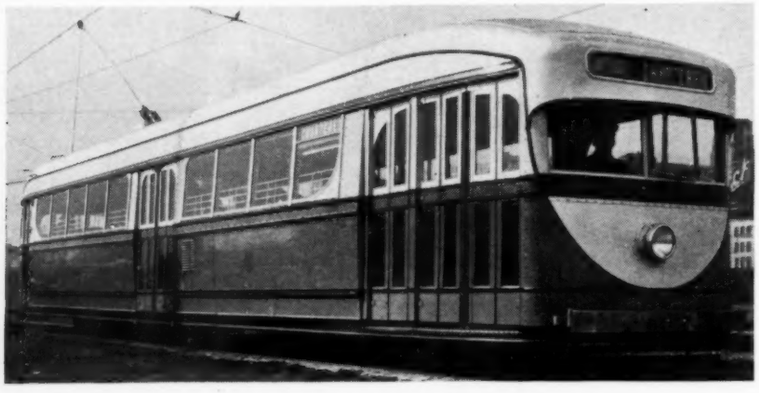
The chief objection to the system as outlined is that for a high outdoor temperature the horsepower per ton of refrigeration would be entirely too high and the hourly output would be greater than in cold weather, which condition would result in very short periods of operation, and consequently "cold 70," that is, a feeling of chilliness even though the room temperature might be 70 F. Certainly some arrangement could be developed to overcome this condition. For instance, assume a 6-cylinder engine connected to three heat-pump pistons. A thermostat placed outdoors could be connected in such a manner as to shut off or turn on the fuel supply to certain of the cylinders of the engine. Inasmuch as the horsepower required is dependent upon the outdoor temperature, this arrangement could be set so that the hourly heat input would not greatly exceed the hourly heating requirements. This would prevent frequent starting and stopping. It might be advisable to have the compressor supplying heat to condenser C_3 in continuous operation, being stopped only by the thermostat in the building. In order to maintain an outlet air temperature of 125 F, it would be necessary to vary the fan speed with the hp output. This could be worked out in conjunction with the operation of the outdoor thermostat or the different cylinders in a number of ways.

Recent experiments in summer cooling and generally accepted methods of computing the maximum cooling load for such a climate as that considered in this article, indicate that cooling equipment should have a heat removal capacity of about 60 per cent of the heat output capacity of the heating plant. Therefore, in this case the capacity of the cooling plant should be 0.6 times 125,000 or 75,000 Btu per hr.

A refrigerating compressor rated on the basis of 86 F on the high side and 5 F on the low side would remove about 18,000 Btu per hr per ton capacity, nominal rating, at the pressures that would exist at the time of maximum cooling. Therefore, the size compressor required would be equal to 75,000 ÷ 18,000 or about 4.2 ton size. At the prevailing pressures the power requirements would be about 1½ hp per ton or a total of about 6.3 hp.

Inasmuch as there would be required for heating an 8.8 ton size compressor and a 10 hp gas engine, it is evident that the cooling load could be easily handled. Probably the best method would be to operate as shown in Fig. 1, with the refrigeration capacity controlled by the gas input to the engine. The efficiency of the gas engine would fall off at less than full load, but the comparative operating cost would, nevertheless, be low.

[Mr. Mowrey's next article will describe other possible arrangements, and will consider briefly some of the practical points involved.—*Editor.*]



More Comfort, Less Noise, Faster Operation

New street car embodying features for faster operation, less noise, more comfort

THE railroads with their high-speed, stream-lined, air-conditioned trains (see page 97, March, 1934, H. P. & A. C.) are not the only members of the transportation industry which are introducing up-to-the-minute equipment for better serving the needs of the traveling public. As a result of three years of research by the presidents' conference committee headed by Dr. C. F. Hirshfeld of Detroit and financed by street-car operating and manufacturing companies, the new car shown here and now in operation on the tracks of the Chicago Surface Lines has been developed. It embodies many features for faster operation, less noise, more comfort.

The heating and ventilation is a combined system. Fresh air is taken into a mixing chamber in the car immediately forward of the center exit door through a louver. This air is mixed with the recirculated air, controlled by a damper. A filter is provided for both fresh and recirculated air.

The air is drawn from the mixing chamber by a two-speed electric blower unit and forced over electric heating coils before entering ducts extending the full length along each side of the car body underneath the seats. These ducts have grilled outlets.

One sliding shutter-type ventilator complete with copper screen is provided in letter panel under sun visor at front end of car. One 1200-watt vestibule heater is installed at front end of car and is controlled by an independent switch near the operator.

Heating · Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

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Gas Heating Equipment: W. E. Stark, *Chairman:* Robert Harper, E. A. Jones, Thomson King, J. F. McIntire, E. L. Tornquist and H. L. Whitelaw.

Heat Requirements of Buildings: D. S. Boyden, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, F. B. Rowley, R. J. J. Tennant and J. H. Walker.

Heat Transfer of Finned Tubes with Forced Air Circulation: F. B. Rowley, *Chairman;* H. F. Bain, H. F. Hutzel, W. G. King, A. P. Kratz, E. J. Lindseth, G. L. Tuve and W. E. Stark.

Minimum Temperature and Method of Introduction of Cooling Air in Classrooms: Perry West, *Chairman;* J. D. Cassell, S. R. Lewis, J. R. McColl, A. J. Nesbitt, G. E. Otis and C.-E. A Winslow.

Oil Burning Devices: H. F. Tapp, *Chairman:* Elliott Harrington, F. B. Howell, J. H. McIlvaine, L. E. Seeley and T. H. Smoot.

Pipe and Tubing (Sizes) Carrying Low Pressure Steam or Hot Water: S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, R. R. Seeber and W. K. Simpson.

Refrigeration in Relation to Air Treatment: A. P. Kratz, *Chairman;* E. A. Brandt, John Everetts, Jr., E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick and J. H. Walker.

Sound in Relation to Heating and Ventilation: V. O. Knudsen, *Chairman;* Carl Ashley, C. A. Booth, F. C. McIntosh, R. F. Norris, C. H. Randolph and J. P. Reis.

Ventilation of Garages and Bus Terminals: E. K. Campbell, *Chairman;* S. H. Downs, T. M. Dugan, E. C. Evans, F. H. Hecht, H. L. Moore and A. H. Sluss.

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Secretary, H. R. ROTH
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Insulating Value of Bright Metallic Surfaces

By **F. B. Rowley,**† ***MEMBER***

Minneapolis, Minn.

This paper is the result of research conducted at the University of Minnesota in cooperation with the A. S. H. V. E. Research Laboratory

THE insulating value of any wall or built-up combination of materials depends upon the thermal properties of its component parts. For the purpose of making a thermal analysis, a wall may usually be divided into sections of homogeneous material, air spaces, and surfaces.

The transfer of heat through homogeneous materials is usually considered to take place by conduction only, although in many cases the transfer is partly by radiation and convection. For instance, in a fibrous insulating material or any material in which there are air spaces, heat is transferred by a combination of conduction, convection, and radiation. In such cases, the structure is usually too complicated to attempt any separation of the total heat transferred into its component paths and the coefficient is considered to be one of conduction only. For exterior surfaces and air spaces, it is often possible and profitable to make a more complete analysis of the heat flow. While such an analysis is often practical, there are many variables and conditions in practice which make the final results only approximate and make check tests desirable.

The purpose of the present discussion is to show by test results the effect of bright metallic surfaces on the heat transmission coefficients, and to point out some of the difficulties in making a theoretical analysis, or in establishing absolute coefficients by test procedure which can be indiscriminately applied to all cases. It is further the object to show reasonable average coefficients which may be used in practice.

Heat is transferred to or from the surface of the material by air contact and by radiation. The amount which is transferred by radiation depends upon several factors one of them being the character of the surface of the material. Until recently, it has seemed sufficiently accurate for practical purposes to consider the materials as having similar radiation characteristics and to use average transmission coefficients where surfaces were involved. Thus such surfaces as paper, wood, plaster, stone, etc., which are used extensively in building construction have emissivity coefficients which vary somewhat and yet which are sufficiently uniform to permit the use of average coefficients insofar as radiation of heat is concerned. The roughness of the surface affects the transfer by convection, but for average conditions this difference has also been neglected in the interests of simplicity in selecting average coefficients. At present, there is an active interest in the use of bright metallic surfaces which have high resistance to the flow of heat by radiation and which give entirely different transmission coefficients than the above mentioned group. In order to meet these conditions, it is necessary to revise the old practice and add new coefficients to the list which will more nearly represent the conditions to be expected when materials are used with low radiating coefficients.

In considering the coefficients for surfaces with high resistance to radiant heat, the same problem exists as for the present common type of building materials. The emissivity varies even for the same material, depending upon the surface finish. For instance, aluminum foil which has a very low emissivity will not always give the same results under test. The polish and character of the surface is not always the same; in fact, under certain adverse conditions the increased insulating value may be entirely lost. If the surface becomes covered with moisture, dust, or some other foreign material which has a high emissivity coefficient, or if it becomes dull and tarnished, its insulating value will be greatly reduced. Due to the difficulties involved in making theoretical calculations, it seems desirable to derive average coefficients from actual tests which may be expected to hold when the materials are built into a wall. With this purpose in view, a series of test results obtained both by the hot plate method and by the hot box method have been analyzed to determine coefficients which may have practical application.

In making the tests, the standard hot plate and hot box apparatus have been used. The general procedure for determining air space coefficients has been to construct a test section with an air space of the desired thickness and lined on one side or both surfaces with the test material. The conductance of the air space has been calculated by standard formula from the test results which have been obtained for the over-all conductance of the built-up section and the conductance of the material exclusive of the air space.

For surface coefficients, the determinations have been by the hot box method only. Since the details of the apparatus, the standard test method, and general procedure have been reported in previous papers, the results for the present tests will, wherever possible, be reported graphically.

The results of individual tests made to determine air space conductance are shown in Fig. 1. These results were taken from several different sources of test data

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For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, June, 1934, Buck Hill Falls, Pa.

and include both the hot plate and hot box methods when using different boundary surfaces, as indicated. The most extended single series of tests are represented by the solid-line curves. The dash-line curve represents the second independent series by the hot plate method in which a different stock of aluminum was used as surface lining. These results agree reasonably well with the first series for similar construction, although the conductances are slightly higher at 0.25 in. and 0.75 in. The test results indicated by points *A, B, C,* and *D* were obtained

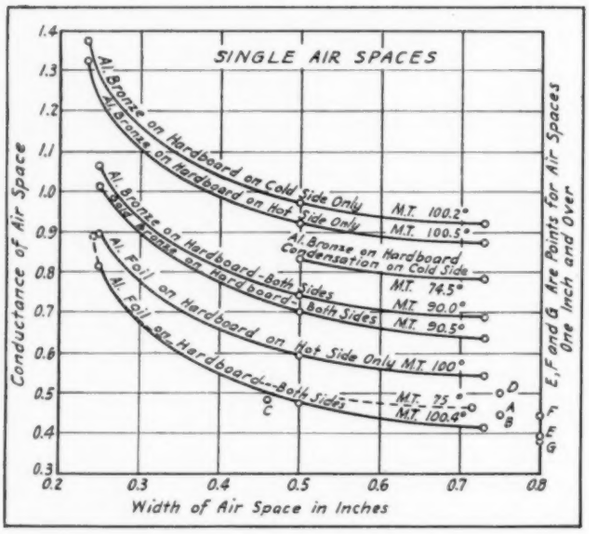
SINGLE AIR SPACES
Conductance of Air Space
Width of Air Space in Inches
Al. Bronze on Hardboard on Cold Side Only M.T. 100.2°
Al. Bronze on Hardboard on Hot Side Only M.T. 100.5°
Al. Bronze on Hardboard Condensation on Cold Side M.T. 74.5°
Al. Bronze on Hardboard—Both Sides M.T. 90.0°
Al. Bronze on Hardboard—Both Sides M.T. 90.5°
Al. Foil on Hardboard on Hot Side Only M.T. 100°
Al. Foil on Hardboard—Both Sides M.T. 75°
M.T. 100.4°
E, F and G Are Points for Air Spaces One Inch and Over

Fig. 1—Results of individual tests to determine air space conductance

from tests made at different times by the hot plate method when using aluminum foil selected from independent stocks. The test results as indicated by points *E, F,* and *G* were all obtained by the hot box method. Test conditions and details for the points indicated are given in Table 1.

While the test results of Fig. 1 show a reasonable agreement between the different series of tests, there is some variation which may be attributed to several causes. In the first place, the surface of the material was probably not in exactly the same condition as selected from different stocks. There is also some possible question as to the size of the test area when comparing the results obtained by the hot plate and the hot box. For those results obtained by the hot plate method, the actual test area was 9 in. square which was divided from the guard ring area by a thin strip of wood. For the tests made by the hot box method, the test areas were 3 ft square. It will be noted that in general those results obtained by the hot box method were slightly lower than those obtained by the hot plate method for specimens of the same width of air space and the same boundary material. The area of test section may have had some bearing on this difference.

For all of the test results by the hot box method, with the exception of point *F,* there were 2 in. x 4 in. studs spaced 16 in. on center, dividing the air space through the test section. For Test *F,* the entire 3 ft square test area was open. Undoubtedly, the presence of the studs through the test section would somewhat reduce the effectiveness of the bright surface.

Taking into consideration all of the test values for air spaces lined on one or both surfaces with aluminum foil as shown in Fig. 1, the average curves for 100 deg mean temperature as shown in Fig. 2 have been constructed.

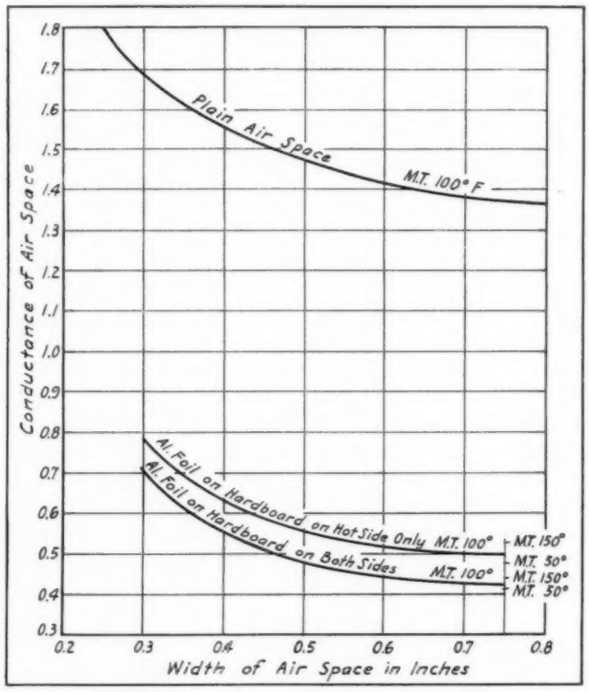
Conductance of Air Space
Width of Air Space in Inches
Plain Air Space M.T. 100° F
Al. Foil on Hardboard on Hot Side Only M.T. 100°
Al. Foil on Hardboard on Both Sides M.T. 100°
M.T. 150°
M.T. 50°
M.T. 150°
M.T. 50°

Fig. 2—Average curves for 100 deg mean temperature

For comparative purposes, the 100 deg mean temperature curve for surfaces lined with fibrous material has been taken from the curves published in the A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, pages 165-168.

From the curves on Fig. 2, a comparison may be made between the emissivity coefficients for the two types of materials by using the well-known Stefan-Boltzmann law and making certain assumptions.

$$(1) \qquad H = 0.172(A)\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right](F_A)\,(F_E)$$

In this formula, the angle factor F_A may be taken as 1, and the emissivity factor F_E may be taken as:

Table 1—Test Values for the Conductance of Air Spaces as Used in Fig. 1

Test Point Fig. 1	Date of Test	Test Method	Thickness Air Space In.	Surface, Hot Side	Surface, Cold Side	Conductance a	Deg F Mean Temp.
A	12/6/'33	Hot plate	0.75	Aluminum foil	Aluminum foil	0.449	75
B	6/13/'32	Hot plate	0.75	Aluminum foil	Wood fiber	0.447	90
C	1932	Hot plate	0.46	Aluminum foil	Aluminum foil	0.485	77
D	5/6/'33	Hot plate	0.75	Aluminum foil	Dull black	0.50	75
E	12/6/'33	Hot box	3½	Aluminum foil	Wood fiber board	0.393	40
F	8/4/'32	Hot box	3	Aluminum foil	Aluminum foil	0.44	40
G	10/15/'30	Hot box	1	Aluminum foil	Aluminum foil	0.383	40

Table 2—Results of Tests by Hot Box Method to Determine Surface Coefficients for a Wall when Covered with Paper and with Aluminum Foil

1 Date of Test	2 Wall. No.	3 Lining for Both Surfaces	4 Mean Temp. of Test	5 Over-all Coefficient, U	6 Conductance Surf. to Surf., C	7 Av. Surface Coefficient	8 Surface Radiation	9 Surface Convection
							Btu	
10/15/'30	78	Paper	40 F	0.691	3.5	1.72	0.733	0.99 —
10/20/'30	78B	Aluminum foil	40 F	0.46	3.5	1.06	0.068	0.99 +

$$(2) \qquad F_E = \frac{1}{\frac{1}{p_1} + \frac{1}{p_2} - 1}$$

in which p_1 and p_2 are the emissivity coefficients for the materials on the corresponding sides of the air space. If the average emissivity coefficient p_1 for the non-reflecting material, such as wood, paper, etc., be taken as 0.9 and it is assumed that the amount of heat given off by convection is the same for both types of surfaces, the emissivity coefficient for aluminum foil surfaces may be calculated from the test results. This method gives an emissivity coefficient of 0.08 which is slightly larger than that often quoted for aluminum.

The mean temperature variations as shown on Fig. 2 are based on the assumption that the difference in heat transfer for various mean temperatures is entirely due to radiation and not to convection. These are calculated and not test values, but since they are small variations, the error can only be slight. The small difference between the curves for single and double air space linings as shown on Fig. 2 is due to the fact that the foil has such a low emissivity coefficient that a single surface intercepts practically all of the radiant heat, leaving but a small portion as a possible saving for the second surface lining. This is readily shown by an inspection of formula (2) which gives the value of the emissivity coefficient.

Substituting emissivity values in this formula,

$$F_E = \frac{1}{\frac{1}{p_1} + \frac{1}{p_2} - 1}$$

The values for F_E are as follows:

No foil on either surface, $F_E = 0.818$
Foil on one surface, $F_E = 0.079$
Foil on both surfaces, $F_E = 0.042$

Thus, $\frac{0.818 - 0.079}{0.818}$ or 90.3 per cent of the radiant heat is stopped by one surface lining and $\frac{0.818 - 0.042}{0.818}$ or 94.8 per cent is stopped by two surfaces, lined.

Referring to Fig. 1, the difference in the values for air spaces lined on one or both surfaces with aluminum bronze is greater than for those air spaces lined on one or both surfaces with aluminum foil. This is due to the fact that the emissivity coefficient for bronze is higher than for aluminum and a single surface lining does not intercept so large a percentage of the radiant heat. It will be noted that the transmission is slightly higher for an air space lined on the cold surface than for one lined on the hot surface. This was found in several cases when using aluminum foil, but in most of these cases condensation was shown to be present on the cold surface. Since the emissivity coefficient for water is high, the presence of a water film on the aluminum surface practically destroys its value.

From the curves of Figs. 1 and 2, it is apparent that the variation in conductance for air spaces of different widths is the same regardless of the surface lining, and the air space must be 0.75 in. or more in width to get the full value of heat resistance. This variation in conductance is due to convection as the radiant heat is substantially the same regardless of thickness.

In considering surface coefficients for building material, the usual procedure has been to consider them as unit coefficients without regard to the percentage transferred by radiation or convection. This appears to be sufficiently accurate for average practice with ordinary building materials. However, for those materials with a low emissivity coefficient there may be an appreciable error, especially for low wind velocity. In most cases, these coefficients have been determined and are used for temperature differences between the surface of the wall and the air, which are substantially the same as the temperature differences between the surfaces of the wall and the surfaces of the surrounding objects, and for average practice the radiant heat may be separated from the convected heat by assuming the same temperature drop for each.

Any radiation coefficient which may be selected for test purposes will not apply to all practical applications, due to the many variations in emissivity coefficients both in the particular wall surface under consideration and in the surfaces of surrounding objects. Average values, however, may be taken for those materials most commonly met with and relations obtained which are sufficiently accurate for practical purposes. Thus it seems reasonable to assume 0.9 as the average emissivity coefficient for the materials which surround the outside of a building wall or for those normal materials which are used in inside construction, with the exception of the special cases of bright metallic or highly reflecting surfaces. Under this assumption, the amount of heat radiated to or from exterior surfaces of walls may be calculated by the Stefan-Boltzmann law with a reasonable degree of accuracy. By using equation (1), the factor F_A may be taken as 1, the emissivity of the wall surface material may be selected as an average value for the particular material from which the wall is constructed, and for normal cases the emissivity coefficient of the surrounding objects may be taken as 0.9. On this basis, corrections can be made to existing wall coefficients.

Table 2 shows the results of tests made by the hot box method to determine the differences in surface coefficients for a wall lined with paper which represents the average building material, and one lined on both surfaces with aluminum foil. In order to eliminate as far as practical all heat resistance in the wall excepting that of the surfaces, a wall was constructed of ⅜-in. thick sheetrock, and the only difference in the two walls was that in Wall No. 78B bright aluminum foil was pasted on to the two surfaces of Wall No. 78.

The surface coefficients in column 7 of Table 2 are the averages for the two surfaces of the wall under test.

Table 3—Reduction in Average Surface Transmission Coefficients Caused by Covering the Surface with Bright Aluminum Foil

Mean Temperature, Deg F	Reduction in Coefficient, Btu
20	0.559
40	0.634
60	0.711
80	0.797
100	0.887
120	0.99
140	1.09

The radiation coefficients in column 8 were calculated from the Stefan-Boltzmann law by assuming emissivity coefficients of 0.9 for the paper surfaced wall and also for the surrounding objects in both cases, and an emissivity coefficient of 0.08 for the aluminum surfaced wall. The heat transmitted by conduction and convection in each case, as shown in column 9, was obtained by taking the difference between the total heat transferred and that transferred by radiation alone. These tests were previously reported in the A. S. H. V. E. Transactions, Volume 38, 1932, p. 491, although the surface coefficients were not analyzed at that time.

The values which have been accepted for surface conductances from building materials may be reduced when using surface finishes which have a high resistance to radiant heat. The reduction will depend upon the emissivity of the surface and upon the mean temperature between the surface and the surrounding objects. It will be independent of air velocity over the surface. For average conditions of aluminum surfaces which are highly polished, it would seem reasonable to make reductions in the transmission coefficients as shown in Table 3 for the different mean temperatures. These reductions, however, cannot be followed blindly, due to the many variables which enter the problem.

In conclusion, it may be said that the general accepted coefficients for wall surfaces and for air spaces which involve surfaces should be revised if special materials with low emissivity coefficients are to be used for surface linings. The reduction will depend upon the surface characteristics of the material. If the emissivity coefficient is known, the correction can be made with reasonable accuracy by calculation. For practical purposes, the test coefficients may be used for aluminum foil lined with air spaces as given in curves of Figs. 1 and 2, and reductions for surfaces as given in Table 3.

This paper is the result of cooperative research work between the American Society of Heating and Ventilating Engineers and the University of Minnesota.

The author wishes to give credit to Mr. William A. Eckley and Mr. Robert M. Lander for the work which they did in obtaining the test values for those points shown on Table 1 by the hot plate method.

Heat Transfer Teaching Course

At the June convention of the *Society for the Promotion of Engineering Education,* to be held at Cornell University, Ithaca, N. Y., there will be a Symposium on Heat Transfer. Five sessions are to be held according to the announcement of Prof. E. R. Queer, Secretary of the Heat Transfer Committee, and the meeting will open at 9:00 a. m., June 19. The subjects of discussion are:

Tuesday—9 a. m. to 12 m.

1. Place and Scope of Heat Transfer Instruction for Undergraduates and Graduate Students in:
   (a) Chemical Engineering.
   Speaker—Prof. A. O. Hougen, University of Wisconsin.
   (b) Mechanical Engineering.
   Speaker—Prof. Ellenwood, Cornell University.
   (c) Electrical Engineering.
   Speaker—Prof. A. D. Moore, University of Michigan.
   (d) Architectural and Civil Engineering.
   Discussion, written preferable.
2. Methods of Teaching Heat Transfer, Theory and Laboratory.
   Speaker—Prof. McAdams, Massachusetts Institute of Technology.
   Discussion. How do others organize and present Heat Transfer Instruction?

Tuesday—2 to 4 p.m.

1. Thermal Radiation—Methods of Investigation and Industrial Applications.
   Speaker—R. H. Heilman, Mellon Institute.
2. Thermal Radiation—Subject Matter and Laboratory Technique.
   Speaker—Prof. H. C. Hottel, Massachusetts Institute of Technology.
3. Conduction—Subject Matter and Laboratory Technique.
   Speaker—Prof. F. B. Rowley, University of Minnesota.
4. Convection—(Natural and Forced)—Subject Matter and Laboratory Technique.
   Speakers—Prof. McAdams and Prof. Drew, Massachusetts Institute of Technology.
   Discussion following each paper.

Tuesday—8 to 10 p.m.

Joint Seminar. Fluid Mechanics, Chem. Eng., Heat Transfer. Application of Fluid Mechanics to Aeronautical Engineering, Chemical Engineering, Civil Engineering and Naval Engineering. See Fluid Mechanics Program.

The Hydrocal: A New Hydrodynamic Instrument for Solving Heat Transfer and Temperature Rise Problems.
Speaker—Prof. A. D. Moore, University of Michigan.

Wednesday—9 a.m. to 12 m.

1. Heat Transfer and the Phenomena of Evaporation.
   Speaker—Prof. W. L. Badger, University of Michigan.
2. Heat Transfer and the Phenomena of Condensation of Pure Condensable Vapors.
   Speaker—Prof. H. Diederichs, Cornell University.
3. Heat Transfer and the Phenomena of Condensation from a Mixture of Condensable and Non-Condensable Gases.
   Speaker—A. P. Colburn, E. I. du Pont de Nemours.
4. The Organization of a Course in Heat Transfer.
   Speaker—E. C. Rack, Johns-Manville Company.
5. What Knowledge of Heat Transfer Does Industry Expect of College Graduates in Mechanical Engineering and Electrical Engineering.
   Speakers—W. J. King and W. L. Knaus, General Electric Company.
   Discussion following each paper.

Wednesday—2 to 4 p.m.

Joint Session—Fluid Mechanics.

1. Dimensional Analysis.
   Speaker—Prof. J. F. D. Smith, Harvard University.
2. Effect of Heat Transfer on Flow of Fluids.
   Speaker—Prof. Drew, Massachusetts Institute of Technology.
3. Economics of Fluid Mechanics and Heat Transfer.
   Discussion following each paper.

Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation

By G. L. Tuve* *(MEMBER)* and C. A. McKeeman† *(NON-MEMBER)*, Cleveland, Ohio

This paper is the result of research conducted at Case School of Applied Science in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

I. Introduction and Summary

HOW does the heat transfer from a forced-convection air heater vary with the air velocity? Is the heat transfer coefficient (per degree of temperature difference) the same for either air heating or air cooling? Are the commercial fin-tube units as effective per square foot of air-side surface as are plain pipe or cast iron sections? What is the best arrangement for securing high heat transfer with a minimum of pressure-drop?

These and many other questions occur to the engineer and should be answered by experimental and research work. While much data are already available, many questions remain to be answered before engineers can predict with confidence the performance of their designs. This paper is a preliminary report on a project in this field, which has been undertaken by the Mechanical Engineering Laboratories of Case School of Applied Science, with the cooperation of the Committee on Research, of the A. S. H. V. E. A study and correlation of available data from the literature are here presented, and experimental results on certain phases of the problem are given.

1. The *major* factors affecting the heat transfer coefficient of finned tubing (or other air heating or cooling units) are three: (a) The design and surface-arrangement of the unit. (b) The mean velocity of the air stream. (c) The character of the air stream, with particular reference to the presence of local eddies or local turbulence. (Surface condensation and evaporation are another major factor in the case of air coolers, but these merit separate study and are not being considered in this paper.)

2. Of *lesser* importance are the diameter of the tubes, the temperature, the kind and the velocity of heating or cooling fluid used in the tubes, and the direction of heat flow, *i. e.*, air-heating or cooling.

3. It is possible to calculate the heat transfer coefficient by a simple equation, but a few tests must be made on each new design-arrangement in order to establish the constants in the equation.

4. The use of flow-disturbers in the air stream immediately preceding a fin-tube section, may increase the heat transfer by 50 per cent or more in certain cases.

5. High velocities are somewhat less effective with extended surface units than with plain tubing, and it is *not* safe to assume that the heat transfer varies as the 0.8 power of the velocity, as it may not vary at a rate greater than the square root of the velocity.

6. There is a definite need for further research in this field (See Sec. IV, *Conclusion*).

II. Heat Transfer Equations

In order to arrive at a method for calculating the heat transfer of a forced-convection air-heating (or cooling) unit,[1] the following variables must be considered:

1. Mean difference in temperature between the air stream and the hot surfaces.
2. The area, shape, size and surface arrangement of the hot surfaces.
3. The mean velocity of the air stream.
4. Character of the air stream as regards turbulence and local eddies.
5. The absolute temperature of the air, with particular reference to the air film next to the heating surfaces.
6. The character of the heating fluid used in the tubes, its velocity, etc.
7. The effectiveness of the extended surface (if any), compared with the prime or direct heat-transfer surface.

Since the surface resistance to heat flow on the outer or air-side is much greater than the similar resistance on the inside of the pipe, it is desirable to compensate for this by increasing the area of the air-side. The simplest extended surface arrangement consists of circular fins of uniform thickness attached to standard round tubing. The effectiveness-factor for the fin surface as compared with the tube surface then depends upon the thickness of fins, depth of fins, distance between fins. and the method of attaching the fins to the prime surface.

It is most convenient to calculate the total heat transfer by the usual simple equation:

$$q = A\,U\,(T_1 - T_2) \qquad (1)$$

where

q = heat transfer in Btu per hour
A = area of air-side surface, square feet
U = overall coefficient of heat transfer per square foot of air-side or external surface, per degree Fahrenheit temperature difference.
$T_1 - T_2$ = mean difference in temperature between the heating fluid and the air. Usually the logarithmic mean temperature difference should be used.

The overall coefficient U may be calculated by combining the inside and outside film conductances and the

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[1] In most of the discussion which follows, an air-heating unit is assumed for convenience. The data and the methods outlined apply as well to air coolers, however, as long as no surface condensation is involved.

conductance of the wall, but since the conductivity of the metal wall is high and its thickness is small, it may be left out of consideration with but little effect on the degree of accuracy. Then:

$$U = \frac{1}{\frac{R}{h_1} + \frac{1}{h_2}} \tag{2}$$

where h_1 is the hot-side and h_2 the air-side film or surface coefficient, and R is the ratio of air-side surface to hot-side surface. This equation is sufficiently accurate for commercial pipe or tubing, but does *not* account for temperature gradients in extended surfaces.

The general equation for the surface coefficient in forced convection, which may be derived by dimensional analysis, is as follows:

$$\frac{hD}{k} = b\left(\frac{DG}{\mu}\right)^n \left(\frac{c\mu}{k}\right)^m = b\,R^n\,S^m \tag{3}$$

where all quantities are expressed in dimensionally consistent units (usually foot—pound—hour), and the symbols are defined as follows:

$\frac{hD}{k}$ = Nusselt number (dimensionless).

$R = \frac{DG}{\mu}$ = Reynolds number (dimensionless).

$S = \frac{c\mu}{k}$ = Stanton number or Prandtl number (dimensionless).

h = film coefficient or surface conductance.

D = diameter (or other linear dimension), feet.

G = mass velocity = linear velocity × density.

c, μ and k are respectively the specific heat at constant pressure, the absolute viscosity and the thermal conductivity of the fluid.

b is the proportionality constant.

This equation, usually ascribed to Professor Nusselt, has been accepted by the leading authorities on heat transfer, and will be found in the standard books in various languages (such as those of McAdams, Schack, Fishenden and Saunders, Grober, etc.) This form of equation should be used when a high degree of accuracy is essential.

For the forced convection of air, the Nusselt equation may usually be simplified, since for moderate ranges of temperature the Stanton number is constant, and the individual values of air conductivity, viscosity and specific heat may even be taken as constant. If this is done, the simplified Nusselt equation becomes:

$$h_2 = \frac{BG^n}{D^{1-n}} \tag{4}$$

where G is the mass velocity, in pounds per hour per square foot and D the diameter in feet. The values of B and n in this equation must be determined by experiment.

For extended-surface units the actual use of Equation 4 is attended with great difficulty because of the temperature gradient of the air-side surface from the base to the periphery of the fin, and because both fin temperatures and adjacent air temperatures are very difficult to measure.

For practical purposes, therefore, these temperature gradients and the relations defined by Equation 2 as well are taken into account by writing a new equation for the overall coefficient, in the form:

$$U = \frac{BG^n}{D^{1-n}} \tag{5}$$

The justification of this form of equation rests entirely on its convenience and on the fact that experimental results are consistent with it.

The values of exponent n and constant B in Equation 5 will *not* be the same as the corresponding values in Equation 4, even if no appreciable temperature gradient exists along the fins. The deviation between the two values of exponent n will not be great, but the exponent of Equation 5 will be smaller.

Recommended Calculation Procedure.—In view of the fact that comprehensive experimental data justifying calculation refinements are not available, the authors recommend that until such data are obtained all forced-convection air heater and air cooler calculations be based on the simple equations (1) and (5), or a combination of the two in the form:

$$q = \frac{ABG^n}{D^{1-n}} \quad (T_1 - T_2) \tag{6}$$

III. Experimental Work at Case School of Applied Science

Evolution of Test Equipment.—Work has been done at Case School of Applied Science on five different pro-

Fig. 1—General view of test duct from inlet end

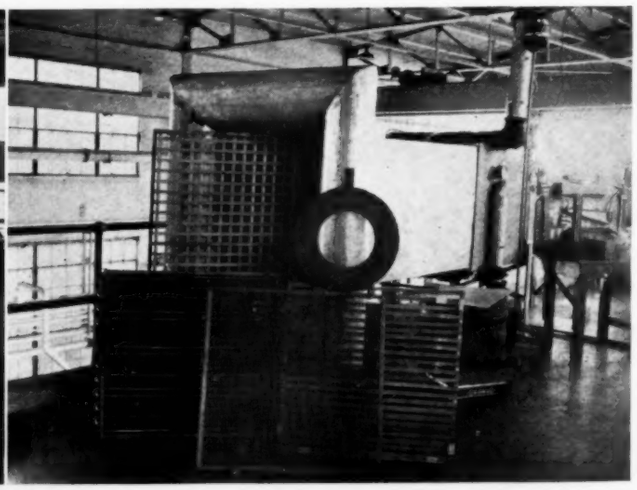


Fig. 2—A group of flow-disturbers and an air-metering orifice

jects, involving as many test arrangements. Both air heating and cooling have been studied, using hot and cold water and steam as heat carriers, and both plain and finned units have been tested. Blow-through and draw-through arrangements have both been used. While several of the variables already mentioned have been studied at some length, the most significant result has been the demonstration of the importance of air-stream conditions, and the results presented herewith are confined to this phase of the subject.

When this work was started, there was no intention of making any special study of air-flow conditions, but it was soon found that these conditions were so important that their effects masked completely the effects produced by certain other variables which the study was to cover.

The first test unit was attached to an air conditioning system, and was tested both as an air heater and an air cooler, using hot and cold water respectively as a circulating medium. With this set-up the variations in heat transfer coefficient with mean air velocity were not as consistent as was desired, and this was found to be due to an unequal distribution of velocities across the face of the test section, accompanied also by changes in the amount of turbulence or local eddying in the air stream as the air velocity was varied. An unequal velocity distribution results in heat transfer coefficients which are low compared with those for uniform velocity across the face of the unit, while local eddies or turbulence result in high coefficients.

To obtain a uniform velocity across the entire test section and an undisturbed or smooth flow, a draw-through unit with a large inlet-bell was decided upon, and four different arrangements were tried, the most satisfactory being that shown in Fig. 1. In these units the air flow was regulated entirely by a variable-speed fan with infinite step rheostats, and to determine the velocity- and temperature-distribution on the upstream side of the test section, traverses were made with an impact tube and micro-manometer and with a thermocouple, respectively. With this arrangement, almost no variation of velocity or air-inlet temperature could be detected.

Tests were next confined to air heating by steam, using first a series-flow arrangement in horizontal tubes, and later a parallel-flow unit with vertical tubes. In all cases, only the tubes projected into the air stream, the return bends or headers being mounted in insulated boxes outside the duct. Low-pressure steam was supplied from either a reducing-valve or a gas-fired boiler with accurate pressure control, and was passed through an electric superheater before it entered the test section. The condensate drain was automatically controlled by using a U-tube waterleg or gravity seal. Continuous vents of ample size were attached at both ends of the condensate header.

In the test runs, after allowing ample time for conditions to become steady, readings were taken every three minutes, during one-half to one hour. Condensate weights were read to 0.01 lb and temperatures to 0.1 F. Air quantities by heat balance were checked by measuring the air flow with a calibrated orifice, and the agreement was within 5 per cent for all runs. Both thermocouples and thermometers were tried for air temperature measurement, but the thermometers, if shielded from radiation, were found to be just as accurate and more convenient. Four readings each of inlet- and outlet-air temperatures were used, but the variation was very small because special mixing devices were used on the outlet side to prevent stratification. The entire test duct and all pipe connections were well insulated, but heat losses were approximated and corrections made therefor. Corrections for radiation from the test section were calculated, but were found to be negligible.

One test unit was constructed in which an elaborate series of fin-temperatures were measured, but more data

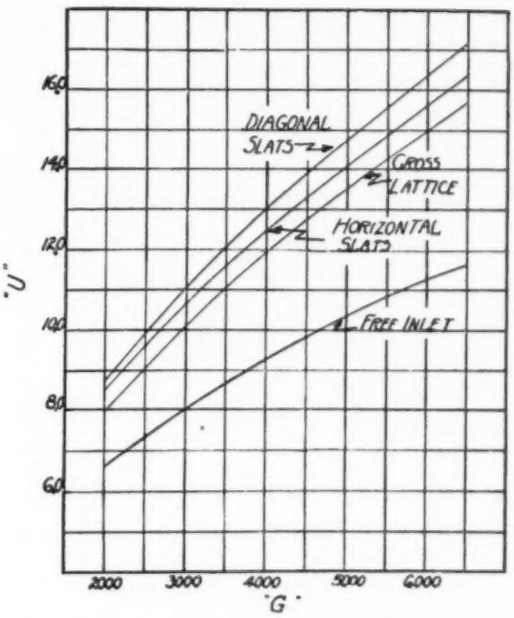


Fig. 3—Effect of flow-disturbers on variation of overall coefficient U with mass-velocity, G

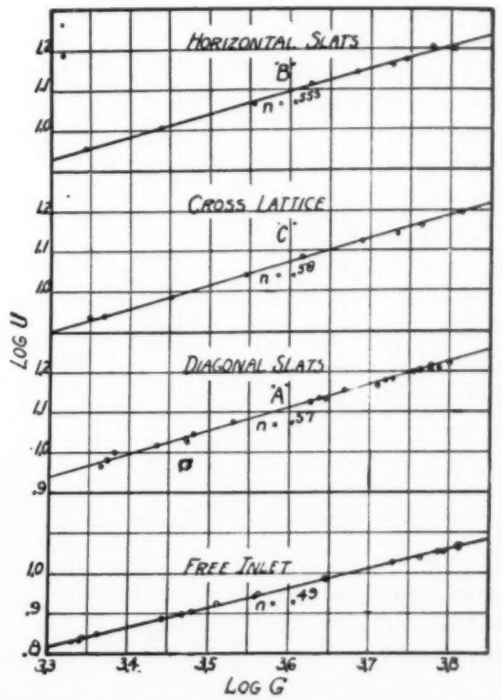


Fig. 4—Curves of Fig. 3 plotted on logarithmic scales, with test points and slopes indicated

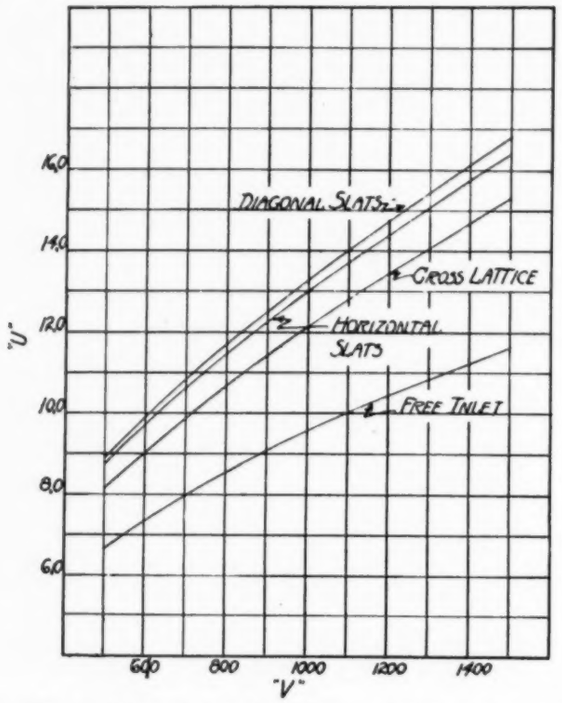


Fig. 5—Effect of flow disturbers on variation of overall coefficient U with mean linear air velocity V

on fin-temperature gradients is to be obtained, and this phase of the subject will be reserved for a later paper.

Nature of Tests.—The tests here reported (Figs. 3 to 6 inclusive), were all made for the purpose of determining the effect of two variables: (1) Mean air velocity, and (2) Air disturbances or local eddy-velocities. Linear velocities from 300 to 1,500 fpm through the free area were covered. Fig. 2 shows some of the grids or flow disturbers which were used in the inlet duct in front of the test section.

These tests were all made on a 35-in. square test section in a single row with ¼-in. clearance between adjacent tubes. Overall diameter of the finned tubing was 1⅜ in. and it was constructed of ⅜-in. smooth copper ribbon 0.012 in. thick spirally wound on ⅝-in. O. D. copper tubing, 7 fins per inch. The assembled unit was tin-dipped for bonding, and its total weight was 0.7 lb per foot.

Tests were made with several types of disturbers or grids having free areas from 25 per cent to 90 per cent of the duct area and spaced at various distances from ½ in. to 8 ft from the upstream face of the test section.

Test Results and Conclusions.—In Fig. 3 are shown the values of the overall coefficient U, (Equation 1), plotted against the mass-velocity of the air G, for three types of flow disturbers or grids, and for free-open inlet or undisturbed flow. The same curves with the test points indicated, are shown separately in Fig. 4. Fig. 5 gives the same data plotted against linear air velocity V, and these curves are shown separately in Fig. 6. All velocities represent the mean air speed through the free or unobstructed area between fins and tubes, *i. e.*, the area considered is the duct area less the projected area of the finned tubing. In these tests the disturbers were mounted one inch in front of the inlet face of the test section.

Fig. 7 shows the pressure-drop losses through the test section including the loss through the disturber-grid.

From these curves and from the supplementary tests, the following conclusions are indicated:

1. For this particular finned tubing the heat transfer coefficient for undisturbed air flow at uniform velocity varies approximately as the square root of the air velocity, as represented by the equation:

$$U = 0.052 \frac{G^{0.43}}{D^{0.51}}$$

2. By producing local eddy-velocities near the test section this heat transfer may be increased, and the increase is greater at high velocities. This is indicated by the equation applying to disturber-grid A:

$$U = 0.045 \frac{G^{0.57}}{D^{0.43}}$$

3. For producing a moderate increase in the capacity of an extended-surface unit, the disturber method may involve a lower

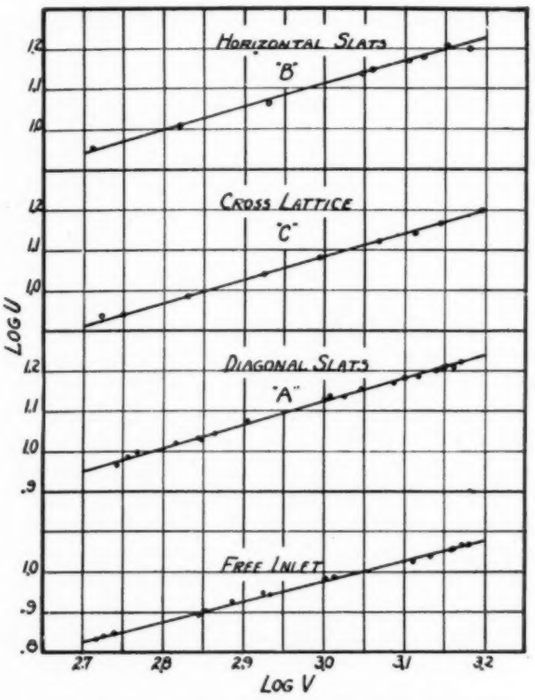


Fig. 6—Curves of Fig. 5 plotted on logarithmic scales, with test points indicated

first cost than either increasing the size of the unit or increasing the air velocity. The method should be useful where air velocities are limited by other conditions, and where a compact unit is desired. The pressure-drop produced by the grid need not be excessive, as shown in Fig. 7.

4. A disturber-grid of about 50 per cent free area is recommended. If the area is further reduced, the pressure-drop increases much more rapidly than the heat transfer. The shape of the elements composing the grids is of less importance. Narrow flat strips set at a 45-deg angle were found to be the most effective. Nozzle-shaped air passages showed little if any reduction in the friction-drop. For best results the air disturber should be placed from ½-in. to 1½-in. from the face of the heat-transfer unit.

Precautions to Be Observed in Using Fin-Tube Units.—The following practical rules are suggested to cover the operating conditions of extended-surface heaters and coolers:

1. Proper distribution of the heating or cooling fluid is very important on account of the high capacity of the fin tubes per foot of tube length. This calls for high velocities, if a liquid is used as the heat carrier, and adequate venting if steam is used. A combination of series and parallel flow is a good arrangement in either case, but if the tubes are operated in parallel with steam, distribution-orifices at the tube inlets may be necessary. In any case the distribution should be checked by feeling the temperature of the fins, if no more accurate method is available.

2. The discharge air from such a unit is highly stratified, and attempts should not be made to measure its temperature without thorough mixing, and even then four or more shielded thermometers should be used.

3. Duct and heat-transfer surface arrangements should be such as to increase the local turbulence as much as possible, though great inequalities in the velocity distribution across the face of the test section are undesirable.

IV. Correlation With Previous Investigations

The main value in experimental results such as those reported above, lies in the extent to which they may supplement or extend the data already available on the subject. A variety of data have been published on the heating and cooling of air by both direct and extended surfaces, but the number and character of the variables involved is such that the effect of each cannot easily be evaluated. As a summary of the available data, Table 1 has been prepared, using Equations 4 and 5 as a basis of analysis.[2] This table gives values for the constants B and n for the calculation of surface coefficients by Equation 4, and overall coefficients by Equation 5. It also gives examples of heat transfer coefficients calculated by each equation, for two air velocities and two pipe diameters. These examples are for heating air by low-pressure steam (the mean air temperature being assumed at 100 F for density calculations).

From Table 1 it is again apparent that the major variables are: (1) Mean air velocity, (2) Air-stream disturbances or local turbulence, and (3) Nature and arrangement of surfaces. These factors, and a fourth factor of temperature level and temperature gradients, will now be discussed in turn.

[2]In view of the excellent bibliography arranged alphabetically in McAdams, *Heat Transmission*, authors' names only are given in this paper and the reader is referred to the above-mentioned bibliography for sources.

Table 1—Summary of Equations for Heating and Cooling of Air by Forced Convection

(Showing Effects of Air Velocity and Pipe Diameter)

Item No.	Equations, Experimenters and Applications (G=pounds/sq ft/hour) (D=feet)	Heat Transfer Coefficients for Heating Air at 100 F, with Pipe Diameters and Velocities of: Diam 1 In. V=500 fpm	Diam 1 In. V=1500 fpm	Diam 2 In. V=500 fpm	Diam 2 In. V=1500 fpm
1	$h=0.0035\ \frac{G^{0.8}}{D^{0.2}}$ Nusselt, Josse, Royds & Campbell For flow inside ¾ In. to 1 In. smooth pipes	2.6	6.4	2.3	5.6
2	$h=0.0085\ G^{0.75}$ Jurges, Taylor and Rehbock For parallel flow over 6 In. to 18 In. square, smooth, flat plates	(No effect of diameter)		1.8	4.1
3	$h=0.0183\ \frac{G^{0.69}}{D^{0.31}}$ Reiher, Carrier & Busey, Rietschel For flow over several rows ⅝ In. to 1¼ In. smooth tubes, staggered	7.8	16.8	6.4	13.7
4	$U=0.049\ \frac{G^{0.585}}{D^{0.415}}$ Commercial bulletin on fin-tubes Flow over 1 row 1⅜ In. O. D. fin-tubes, ⅝ In. fins, 7 per inch	12.2	23.1	8.5	17.7
5	$U=0.045\ \frac{G^{0.57}}{D^{0.43}}$ Authors, Case School Research Flow over 1 row 1⅜ In. O. D. fin-tubes, ¾ In. fins, 7 per inch, with flow disturber of diagonal slats	10.4	19.4	7.7	14.3
6	$h=0.033\ \frac{G^{0.56}}{D^{0.44}}$ Hughes, Reiher, Vornehm, Gibson Flow over single tubes or pipes, ¼ In. to 2 In. O. D.	6.7	13.3	5.1	10.0
7	$U=0.052\ \frac{G^{0.49}}{D^{0.51}}$ Authors, Case School Research Flow over 1 row 1¾ In. O. D. fin-tubes, ⅝ In. fins, 7 per inch, with smooth flow-stream	7.9	13.4	5.5	9.4
8	$h=0.061\ \frac{G^{0.51}}{D^{0.49}}$ T. E. Schmidt[a] Flow over 6 rows 7 In. O. D. fin-tubes, 2 In. fins, 15 per foot	9.0	15.7	7.3	12.7
9	$h=0.044\ \frac{G^{0.4}}{D^{0.6}}$ L. V. King Flow over very small wires	100 for diameter 0.005 In. and 500 fpm velocity			

[a]For curves showing Schmidt's results, see article by King and Knaus, *Mechanical Engineering*, May, 1934. This article also gives other curves showing similar data to above table.

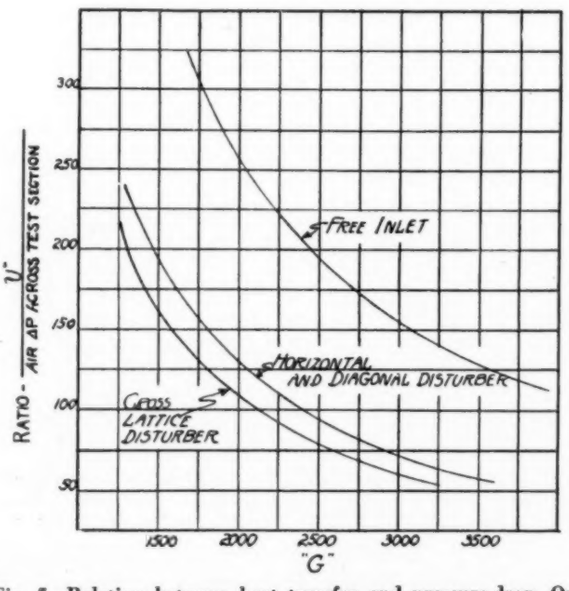


Fig. 7—Relation between heat transfer and pressure-drop. Ordinates are Btu per hour per square foot per degree difference between steam and air, per inch of pressure-drop through test section and disturber grid

Air Velocity and the Velocity Exponent n.—The value of the exponent n in the heat transfer equations for either surface coefficient or overall coefficient (Equations 3, 4, and 5,) is in the case of any given section a direct index of the effect of air velocity on the heat transfer. Table 1 shows that this exponent varies from 0.4 to 0.8, and Table 2 has been prepared to indicate the importance of this variation in practical calculations. Thus if the heat transfer is 10 at a velocity of 200 fpm, and it varies as the 0.4 power of the air velocity, it will have reached 25 at 2000 fpm, but if the heat transfer varies as the 0.8 power of the air velocity, it will increase from 10 at 200 fpm to 63 at 2000 fpm.

In any specific case the value of n is dependent on three things: (1) The diameter, shape and smoothness of the heat-transfer surface. (2) The character of the air stream. (3) The temperature-gradients along the

fins or extended surface, if any. (This last statement applies to Equations 5 and 6 only).

Table 2—Effect of the Velocity Exponent *n*

(See Equations 4, 5 and 6)

Values of Exponent *n*	Relative Heat Transfer (Surface Conductance) for Velocities of—		
	200 fpm	500 fpm	2000 fpm
$n = 0.4$	10	14.4	25.1
$n = 0.5$	10	15.8	31.6
$n = 0.6$	10	17.3	39.8
$n = 0.7$	10	19.0	50.0
$n = 0.8$	10	20.8	63.0

As regards the actual value of the velocity exponent for finned tubing and other extended surface units, McAdams states that it varies from 0.58 to 0.74 (*Heat Transmission,* page 233) and an analysis of the performance tables published by five American manufacturers shows values covering approximately this range.

Effect of Air-Stream Disturbance.—Unfortunately we have no satisfactory means for observing or measuring the smoothness of an air stream. We may state that Reynolds number is a criterion of turbulence, but this tells only a small part of the story. Noise furnishes some index of stream conditions, but there are many other sources of noise in a system. Photographs of smoke flowing in the air stream are illuminating but they are difficult to get and are successful only over a small range of velocities. Friction or pressure-drop furnishes a good index, but only in case the system remains geometrically similar. The hot-wire anemometer has possibilities for turbulence measurement, but this application is only in an experimental stage.

It thus becomes necessary to make a mere estimate of the degree of flow disturbance from a knowledge of the operating conditions. Departures from smooth-stream conditions are a function of the geometrical shape and surface smoothness of the objects or walls over which a fluid is flowing. Local eddies or vortices may be created, and persist in the stream for some distance past the point of their formation. Apparently anything which will assist the moving air in scrubbing off the viscous or stagnant air film adhering to the heating surface, increases the heat transfer by bringing a larger percentage of the total air particles into close proximity with the hot surfaces. In many cases of stream disturbance, high local velocities exist, and these may play a major part in increasing surface conductance.

Most efforts to produce disturbed air conditions have accomplished this by the shape and arrangement of the heat transfer surfaces with respect to the air stream. Reiher's data on 5 rows of plain 1-in. pipes showed about 28 per cent greater heat transfer for the staggered arrangement than for the in-line arrangement. Similar results were reported earlier by Carrier and Busey. Rowley, Jurges, Ott and others have reported increases in surface coefficients of 5 per cent to 30 per cent with rough plane surfaces as compared to smooth ones. Rowley found that for a flat plate mounted in a wind of 15 to 25 mph velocity, the surface coefficient was increased in varying amounts, up to about 25 per cent as the plate was placed at various angles with the stream from 0 deg to 90 deg. Even larger increases were reported by Reiher. Schey and Bierman obtained an increase of about 50 per cent in the heat transfer of a finned airplane engine cylinder at flying speeds, when it was placed at 45 deg with the air stream as compared with either the parallel or the 90 deg positions. Reiher and Hughes both demonstrated that the heat transfer from a cylinder in an air stream is increased by placing strips or tubes in front of it.

Corrugated-plate heat exchangers have shown a larger coefficient than flat-plate exchangers. Baffles and helical guide-shapes have been used inside pipes to increase heat transfer. Many types of baffles are used in shell-and-tube exchangers for liquids. Screens and decorative grilles are used in front of automotive radiators.

But it is difficult to classify these flow disturbers and little has been published regarding the quantitative evaluation of their effects at various air velocities. There is no doubt but that such disturbers are more effective at high fluid velocities than at low ones. The results shown in Fig. 5 are conclusive on this point.

Friction loss or pressure drop is important in determining the power required to move the air stream and hence the relation between increase of heat transfer and increase in friction loss is an important one. In the case of two disturbers which have the same effect on the heat transfer, the one may produce much less friction loss than the other. Colburn and King demonstrated that by breaking up the flow of air through a tube by packing it with granular material, the heat transfer could be increased as much as six times, but the friction loss was thereby increased 200 times. They concluded that the coefficient might to better advantage be increased by using a higher velocity in a plain tube.

Friction losses are of two kinds, skin friction and eddies or turbulent friction. While much of the data given in this paper indicate a close relation between local turbulence and heat transfer, there is also evidence that the latter is related to skin friction. Hughes found greater heat transfer per unit area for a streamline section than for a cylinder, and from this Fishenden and Saunders conclude, "it appears that heat transfer is directly related only to skin friction."

Effect of Temperature.—Temperature effects are of two kinds, those due to the absolute temperature level and those due to temperature differences or temperature gradients.

The surface coefficients are affected somewhat by the absolute temperature since the viscosity (μ), the conductivity (k) and the specific heat (c) of the fluid are affected, (see Equation 3). To illustrate the magnitude of these effects, examples of typical cases of air heating and air cooling are included in Table 3 (see *Conclusion*).

Table 3—Example of Differences Due to Temperature Level

(*Calculated for 1½-in. O.D. fin-tube unit, same unit in all cases.*)

Temperature Conditions	Air-Surface Heat Transfer Coefficients at Velocities of—	
	500 fpm	1500 fpm
HEATING: Typical air-heater using steam at 5 lb gage	7.0	13.5
COOLING: Typical air-cooler using high-velocity water circulation at 40 F	6.5	12.5

While the difference due to temperature level is large enough to be worth taking into account in accurate heat transfer calculations, it is smaller than the uncertainty which usually exists regarding the air-stream conditions.

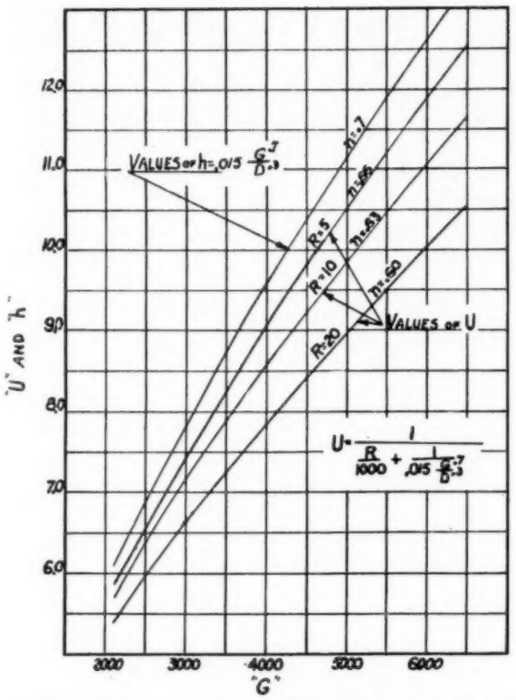


Fig. 8—Variation of overall coefficient of a fin-tube unit with ratio of air-side surface to steam-side surface; no temperature gradient in fins

The effect of temperature differences and temperature gradients is discussed in the next section.

Effect of Nature and Arrangement of Surfaces.—From Equation 3 the effect of tube diameter is shown to be proportional to D^{1-n}, and this relation has been confirmed by experiment for plain pipes and tubes. There are little or no data on this point with respect to finned tubing. The constants given in items 5, 7 and 8 of Table 1 have been derived on the assumption that the diameter should enter in the above manner for finned tubing also, and that the effective diameter is the overall diameter.

The effect of surface arrangement on the value of the velocity exponent n (because such arrangement affects the nature of the air flow), has been discussed, but the presence of extended surfaces affects the value of n in two other important ways. With thin fins, an increase of air velocity tends to bring the temperature of the fin periphery closer to the air temperature, and to set up a temperature gradient or difference between the prime surface and the extended surface. This makes the extended surface less effective and reduces the value of n in the equation for overall coefficient, (Equation 5 or Equation 6).

But even if there is no temperature gradient in the fins, the mere existence of extended surfaces changes the value of n, as may be proven by substituting values in Equation 2. Fig. 8 shows this effect. Taking as an example a steam-heated fin tube with a 10 to 1 ratio of air-side to steam-side surface, (*i.e.*, $R = 10$), assume that the steam-side coefficient $h_1 = 1000$, and that the air-side surface coefficient is represented by the equation: $h_2 = 0.015\ \frac{G^{0.7}}{D^{0.3}}$. Calculating values of U by Equation 2 and plotting them in turn against G, we have a new curve (Fig. 8) which indicates that U varies approximately as $G^{0.63}$. Other area ratios (values of R) give the other curves of Fig. 8, all other assumptions remaining the same.

Conclusion: Thus it is evident that while convenient methods of calculation are available after the constants for a particular design have been established by experiment, yet much remains to be learned about the effect of such primary variables as tube and fin diameters, fin thickness and fin spacing. We know a little about the practical effect of air-flow conditions, but we lack any method of adequately measuring these conditions in a given installation and the theory or principles underlying these effects have not been clarified. We may state that the heat transfer coefficient varies directly with some power of the velocity, but whether this variation is as the square root or as the 0.8 power, or some value between, we can only predict after further work has been done on the effect of the primary design proportions. A rough classification of air heaters and coolers, with suggested heat-transfer equations and examples of the results obtained by their use are given in Table 4, but these values should be used only for approximations. For accurate results, a few test-runs on each design of unit are necessary.

Table 4—Recommended Heat Transfer Equations for Approximate Calculations

	Applications	Equation
Class I.	Large plain tubes, staggered arrangement or with flow disturbers.	$h = \frac{0.22\ G^{0.7}}{D^{0.3}}$
Class II.	Common large fin-tubes, staggered arrangement or with flow disturbers.	$h = \frac{0.04\ G^{0.6}}{D^{0.4}}$
Class III.	Small tubes with large fins, or small plain tubes widely spaced; smooth air flow.	$h = 0.05\sqrt{\frac{G}{D}}$

Equations apply to both air heaters and coolers. h = Btu per hour per square foot of total air-side surface per degree difference between mean air temperature and primary tube-surface tmperature. D is in feet. G is mass velocity in pounds per square foot per hour.

Western Michigan

March 12, 1934. The Western Michigan Chapter convened at the Union Building, Michigan State College, East Lansing, Mich., continuing its practice of meeting in various parts of the section embraced by the Chapter. Forty members and guests attended the dinner and the technical session following.

There were two speakers, both members of the faculty of Michigan State College, who reviewed several technical papers presented at the 40th Annual Meeting of the Society in New York City. Prof. H. B. Dirks, dean of the Mechanical Engineering College, gave a resumé of Thermodynamic Properties of Moist Air, by Prof. J. A. Goff, and reviewed the fundamental laws and properties of both dry air and water vapor. This was followed by a discussion of the application of the formulae given in Professor Goff's paper with the resultant tables and their application to practical problems, illustrating their value.

Prof. L. G. Miller, Department of Mechanical Engineering, summarized the papers, Air Conditioning in its Relation to Human Welfare, by Dr. C. A. Mills; Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten; Comfort Cooling with Attic Ventilating Fans, by G. B. Helmrich and G. H. Tuttle; and Study of Summer Cooling in the Research Residence for the Summer of 1933, by A. P. Kratz and S. Konzo.

These papers took on an additional value by the interpretations and illustrations given by the speakers. An interesting open discussion followed and O. D. Marshall reports that undoubtedly this would have continued late into the night had not many members had to drive to distant points in the state.

The Air Conditioning System of the New Metropolitan Building—First Summer's Experience

By **W. J. McConnell, M. D.,*** ***(NON-MEMBER)*** **and I. B. Kagey**** ***(MEMBER)***

New York, N. Y.

THE new home office unit building of the Metropolitan Life Insurance Co. in New York City, Fig. 1, is the largest completely air conditioned office building in the world. The building is 28 stories in height, with four basements, two of which are conditioned; having a floor area of 23 acres and a capacity of 15,000 000 cu ft, the entire office sections and lunch areas being completely air conditioned.

The system was started in December, 1932 and the period of summer operation began in May and continued until late in October, 1933. It was considered undesirable to attempt to maintain any constant temperature during the summer months, in order to avoid too great a contrast between the inside and outside temperatures. The outdoor temperature on any particular day determined largely the temperature to be maintained indoors. The inside temperature approached the outside temperature as the latter approached 80 F. The maximum temperature maintained indoors reached 79 when the outdoor temperature was above 90. The average range of indoor temperature for the summer months extended between 74 and 79 F with a range of relative humidity between 43 and 53 per cent.

On entering the conditioned area during the warm weather one did not experience the sensation of chillness, but, rather, welcomed the cool environment. One was able to work through a full-day period without the discomfort and depression formerly suffered during hot weather. It might be assumed generally that those working in a cool environment felt less fatigued at the end of the day. On leaving the building at the end of the day, the heat of the outdoor air was intense at first but the effect was not lasting.

The results of exposure of approximately 6000 employees to the conditions maintained during the summer months have been, from the viewpoint of comfort, entirely satisfactory. What effects such exposure has on the health of these employees so far have not been determined. Such health correlations in the past as have seemed significant are usually open to criticism for want of adequate records of conditions, or because diagnostic data were lacking to support the findings. Careful records of causes of absenteeism of all employees are made and it is hoped that an analysis of these records after a period of three or four years may assist in appraising the value of controlled air environment in relation to health.

Although dust and bacteria counts were not determined during the summer months, a dust count was made during the month of December in one of the occupied sections. The standard procedure of the United States Public Health Service, using the Greenburg-Smith impinger, was followed. The count amounted to

Fig. 1—Air conditioned office building of Metropolitan Life Insurance Co.

1.2 million particles per cubic foot of air in contrast to 2.9 million particles in an occupied section of an adjoining non-conditioned building. It is realized, however, that many more counts, including bacteria, should be made before any conclusions relative to the dust concentrations in the respective buildings are submitted.

While the nature of the dust encountered determines its harmfulness when inhaled or ingested, it must be remembered that dust also carries and spreads bacteria, which may be the origin of many respiratory diseases. Leonard Hill contends that the dusty, warm atmosphere of crowded rooms not only spreads infection but lessens the defense of the membrane which lines the breathing passage.

Some individuals are peculiarly sensitive to dust containing traces of foreign material, such as pollen and molds, and suffer from hay fever, asthma, and bronchitis

*Assistant Medical Director.
**Air Conditioning Engineer.
Presented at the 40th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York, N. Y., February, 1934.

Table 1—Metropolitan Life Insurance Company—Floor temperature and humidity—August 1, 1933

	North				East				South				Avg.			
	D.B	W.B.	D.P.	%	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%	D.B.	W.B.	D.P.	%
1st Floor	76.2	63.3	55	49	76	63	55	48	80	65.2	56.5	45	77.4	63.8	55.5	47.3
1st Mez.	77	63.5	55	47	77	63.5	55	47	76.5	63.5	55.2	48	76.8	63.5	55	47.3
2nd Floor	77	63.5	55	47	78.5	64.5	56	46	77.5	65	57	50	77.7	64.3	56	47
2nd Mez.	76.5	63.5	55.2	48	77.8	64	55.6	46	78.5	64.8	54.8	44	77.6	64.1	55.2	46
3rd Floor	77.5	64.3	56.5	48	76.8	63.5	55	48	76	63.2	55.2	49	76.8	63.7	55.6	48.3
3rd Mez.	76.5	63.5	55.2	48	77.3	65	57	50	76.8	63.8	55.5	48	76.8	64.1	55.9	48.7
4th Floor	77.5	63.5	55	45	79	65	56.5	46	76.2	62.8	54.5	46	77.6	63.8	55.3	45.7
4th Mez.	76.5	63.5	55.2	48	77.8	64	55.6	46	77	63.5	55	47	77.1	64.5	55.3	47.7
5th Floor	78	64	55.5	46	79	65	56.5	46	78.8	64.2	55	44	78.6	64.4	55.9	45.3
6th Floor	79	65	56.5	46	78.5	64.5	55.3	44	76.8	63.2	54.8	47	78.1	64.2	55.6	45.7
7th Floor	79	65	56.5	46	78.2	63.8	55	44	76	61.5	52.5	46	77.4	63.4	54.9	45.3
8th Floor	79.5	65.5	57.2	47	77.5	63.4	54.6	45	77.2	63.5	55.2	47	78.1	64.1	55.7	46.3
9th Floor					78.5	64.5	56	46	78.5	64.5	56	46	78.5	64.5	56	46
10th Floor	78.6	65.3	57.5	49	77.5	64	55.2	46	77.5	63.8	55.2	46	77.9	64.4	55.4	47
11th Floor	78.5	65	57	48	78.5	64.5	56	46	78.5	64.5	56	46	78.5	64.7	56.3	46.9
12th Floor	77	64	55.5	44	77.5	63.7	55.2	46	78.5	64	55.5	44	78.2	63.9	55.4	44.7
13th Floor	77	63.5	55	47	76.5	63.5	55.2	48	78.5	64.5	56	46	77.3	63.8	55.4	47
14th Floor	78	63.5	55	47	77.5	63.8	55.2	46	80.2	65.2	56.3	44	78.2	64.2	55.5	45.7
16th Floor	79	62.5	53	42	78.5	63.5	54	43	78	63	53.5	43	78.2	63	53.3	42.7
17th Floor	78	64	55	43	79	64	55	43	79	64	55	43	79	64	55	43
18th Floor	76	63	63.5	43	78	63	53.5	43	77.5	63	54	44	77.8	63	53.5	43.3
19th Floor	78	62.5	54	46	77	63	54	45	77.5	63.5	55	45	76.8	63	54.3	45.3
20th Floor	76	63.5	55	45	78	64.5	56.5	48	79	65	56.5	46	78.3	64.3	55.7	46.3
21st Floor	77	62	53	45	77	63	54	45	76.5	63	55	47	76.5	62.7	54	45.7
22nd Floor	79	63	54	45	79	63.5	54.5	42	77.5	63	54	44	77.8	63.2	54.2	43.7
23rd Floor	79	64	55	43	78	63	53.5	43	77	63	54	45	78	63.3	54.2	43.7
24th Floor	78	64	55	43	78.3	63.3	54	43	78	63.5	54	44	78.4	63.6	54.5	43.3
25th Floor		63.5	54	44	78	64	53.5	46	78.5	63.5	54	43	78.2	63.7	53.8	44.3
26th Floor 1									79	62.5	51.5	39				
26th Floor 2									76.5	62	52.5	43				
26th Floor 3					74.5	63	56	52								
26th Floor 4					77	63.5	55.5	47								
26th Floor 5	77	63	53.5	44												
26th Floor 6	78.5	66	59	52												
2nd Basement	77	66	59.5	56	77.5	66	59	54	77	66	56.5	56	77.2	66	59.3	55.3
3rd Basement					78	65	57.5	50	80	67	59.5	50	79	66	58	50

Readings by Operating Staff, Metropolitan Life Ins. Co., 11 a. m. to 3 p. m.

through breathing such dust. These individuals are greatly relieved so long as they breathe dust-free air.

That greater efficiency follows exposure of the body to environmental conditions which permit loss of excessive body heat, in contrast to a hot atmosphere, has been repeatedly demonstrated. Where the contrast is not so great, as for example, between offices not cooled during the summer months and offices maintained at a comfortable temperature, the increased efficiency of those working under comfortable conditions is difficult to measure with any degree of accuracy. While the authors are convinced that efficiency is greater, no practical method of determination has been devised.

The system was operated eleven hours a day, except in extremely hot weather, when the system was started two hours instead of one hour before the office opened. Tables 1 and 2 give some of the results which were recorded during one of the hottest days in the summer of 1933. Fig. 2 gives a record of the outside wet and dry bulb temperature for a 24-hour day. Table 1 is a thermal record for the entire building between 11 a.m. and 3 p.m. Each floor that is conditioned, is divided into three exposures: north, east, and south; and the average dry bulb and wet bulb temperature, with the corresponding dew point and relative humidity, is indicated for each exposure. The average of the three exposures for each conditioned floor, except the 26th floor which is divided into various laboratories, is indicated under the heading average.

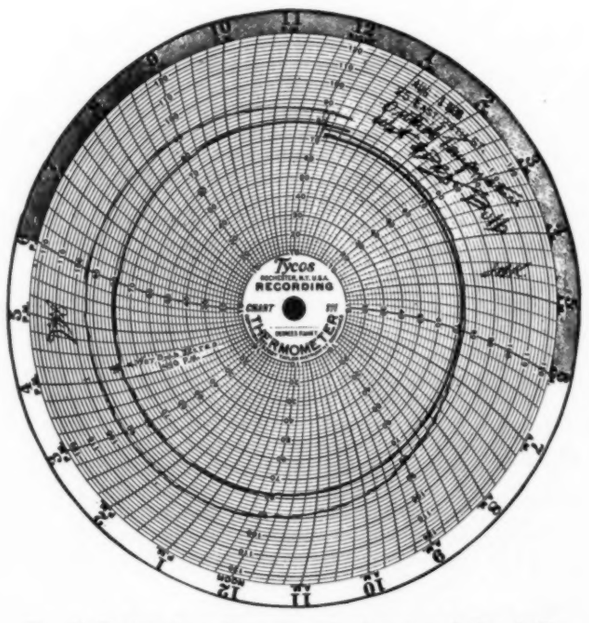


Fig. 2—Twenty-four hour record of wet and dry bulb temperatures

Table 2 is an outline of the results of a refrigeration test made at noon when the lunch rooms were occupied, thus demanding a maximum building load. On this particular day the system was started at 6 a.m., two hours before the personnel entered the building. On the day preceding, the outside temperature averaged 90 deg dry bulb and 74 deg wet bulb, and the system was shut down at 6 p.m. as usual.

It was found necessary to operate the refrigeration equipment for a total of 1230 hours over a period of eight months, as shown in the following tabulation:

Month	Days Refrigeration	Hours Refrigeration
April	3	31
May	16	175
June	21	226
July	20	238
August	23	273
September	20	205
October	8	62
November	3	20
Total	114	1230

If the building had operated 5½ days a week instead of 5 days it would have been necessary to operate the refrigeration for 110 additional hours or 22 half days, bringing the total up to 136 days or 1340 hours when refrigeration would be necessary.

During the 1230 hours of operation the equivalent of

29,000 tons of melting ice was produced, which would represent approximately 1,000,000 cu ft of ice.

Table 2—Metropolitan Life Insurance Company—Refrigeration capacity test

August 1, 1933

Outside temp 91 deg D.B.—78.5 deg W.B.
Four machines full speed

Water entering condensers	66 F
Water leaving condensers	87 F
Water entering coolers	49.5 F
Water leaving coolers	40.0 F
Temp drop in coolers	9.5 F
Gpm through coolers	3420
Heat absorbed per hr	16245000 Btu
Tons of refrigeration	1353.7
Tons of refrigeration to office space	1125.7
Tons of refrigeration to restaurants	228.0
Kw in-put	1222
Hp in-put	1635
Hp in-put per ton	1.21
Gpm condenser water	1880
Gpm condenser water per ton	1.39

Very definite costs as to operation, maintenance, repairs and additions are kept on the entire system. However, the data are not complete for the past year's operation due to the lack of sufficient metering equipment prior to July.

The air conditioning plant is divided into two main systems, as illustrated in the diagrammatic sketch Fig. 3. One supplies the office portion of the building which consists of the first to the twenty-sixth floors inclusive, and the other supplies the lunchrooms located in the second and third subbasements. The former consists primarily of eight dehumidified air fans and four large dehumidifiers which provide air for fifty booster fans or recirculatory units. The dehumidified air fans and dehumidifiers (humidifiers in winter) are located on the fifteenth floor which is the central point of distribution (Section B-Fig. 3.) The supply of outside air is received on this floor through louvers in the building's north and east walls. The outside air is mixed with a proportionate amount of air returned from the various floors and in turn passes through the dehumidifiers where it is properly conditioned. After leaving the dehumidifiers in an almost saturated state, the air is forced to the upper and lower floors through galvanized iron shafts which are cork insulated. There are four main supply shafts, two of which supply the northerly and two the southerly halves of the office areas of the building with conditioned air. There are two booster or recirculatory fan units located in each of the conditioned floors which derive their supply of conditioned air from the north and south main shafts, respectively. Each unit draws a proportionate amount of air from the office area it supplies and this air, in turn, after mixing with the conditioned air supplied to the unit, is delivered to the office area. During the winter the humidified air is supplied from the shaft and after mixing with the air drawn from the office area is passed through small booster heating coils which temper the air to 70 F before it is discharged into the office area. The heat losses in the office areas are offset by means of radiators located under each window.

Combination return and exhaust fans (Fig. 3, sketch B) are located on the fifteenth floor and are connected at this level with four return shafts installed alongside the four main supply shafts. These fans draw from the shafts air exhausted from each floor and either deliver it to the dehumidifier or exhaust it to the outside.

The method of introducing the conditioned air into the office areas is of interest because the method adapts

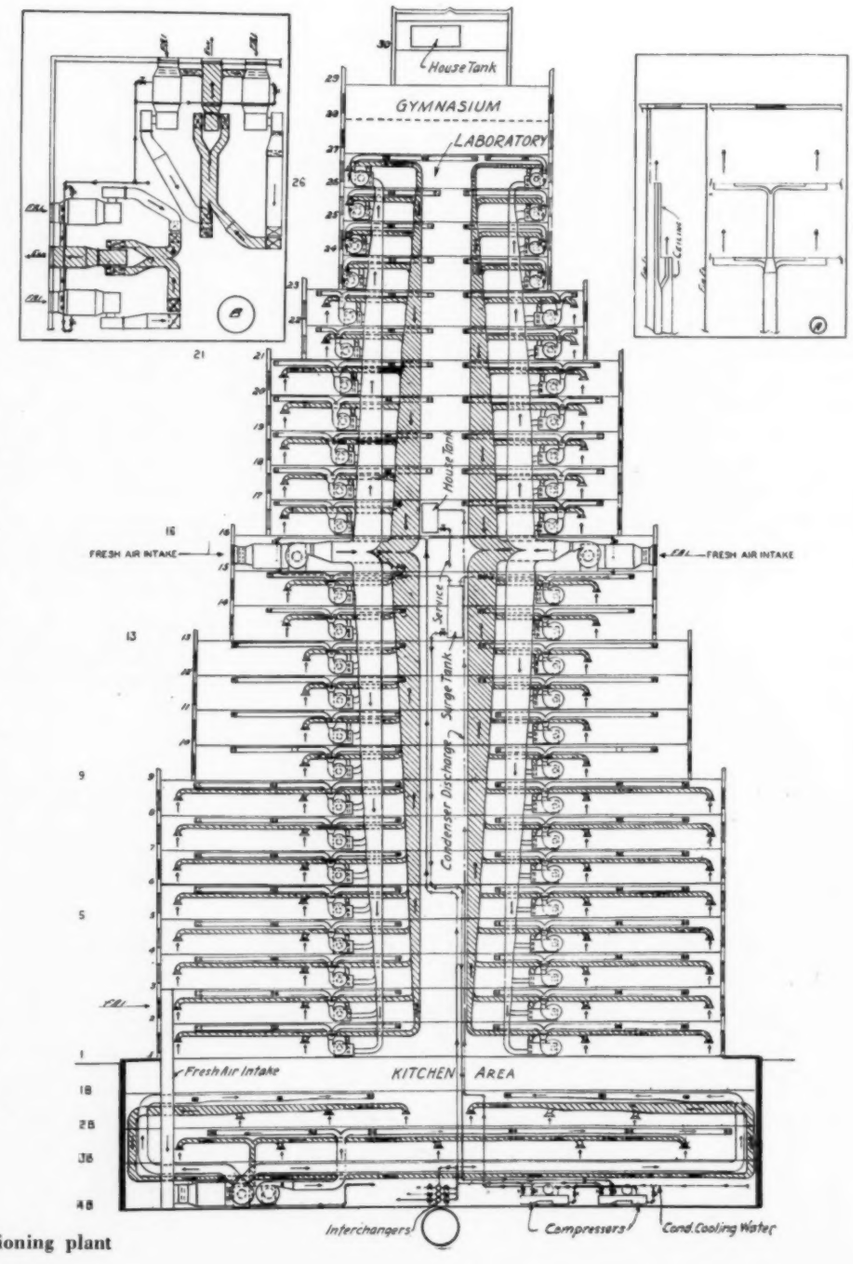


Fig. 3—Diagrammatic sketch of air conditioning plant

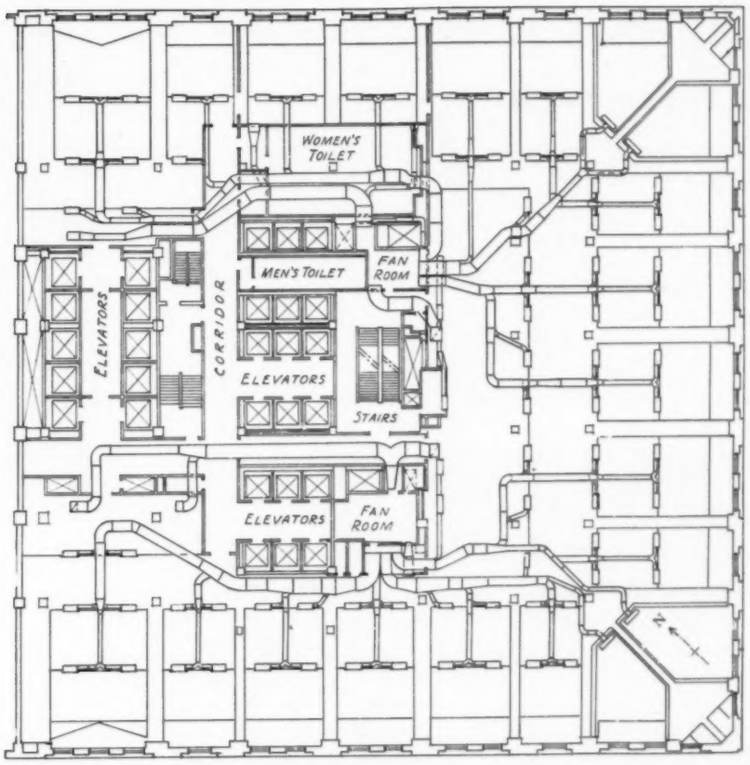


Fig. 4—Floor plan showing typical duct layout

itself both to architectural appearance and to uniform distribution. The office areas generally are large open spaces having very few partitions as indicated in Fig. 5. The large surface of ceiling is divided into successive steps which radiate upward and outward from the service area of the floors to the windows on three sides of the building. The average ceiling has four such steps each about 7 in. above the preceding step. The openings from the supply outlets are located on the face of these steps and concealed by specially designed strip metal grilles 4 ft in length and 4½ in. in height. (Fig. 3 sketch A).

Careful study was given to the design of the ducts concealed in the ceiling and leading from the two respective fan rooms to the supply grilles in order to maintain maximum ceiling height. The duct system consists of numerous branches having an average depth of 7 in. and in some instances these branch ducts pass through openings cut out in the steel girders to permit the ducts to reach extreme corners of the office areas. A typical floor plan showing the duct layout is shown in Fig. 4.

The air is exhausted from the office areas through large registers located in the ceiling soffit. There are on an average ten return registers per floor and as there are few partitions, these registers are situated adjacent to the inside service walls, thus eliminating the necessity of an extensive return duct system which would occupy valuable head room.

The dehumidifying equipment for the lunch rooms is located in the fourth subbasement and draws its supply of outdoor air through a shaft leading to stone grilles in the north wall at the second floor level.

This system, which is the second of two main systems, consists of three dehumidifiers, two of which supply the lunch room in the second subbasement and the third supplies the lunch room in the third subbasement.

Fig. 5—An office area showing method of air distribution

Fig. 6—Lunch room area showing air distribution

Unlike the first main system previously described, this system recirculates and returns the air at each dehumidifier thus requiring only three fans.

The distribution of air to these areas is effected in a different way from the method used in the office areas on account of the different type of ceiling. In these areas the ceilings are flat and the air is introduced into the rooms through one foot square openings in the ceiling. Suspended directly under these openings and four inches below the ceiling are solid plates three feet square which serve as baffles for the distribution of air as shown by Fig. 6. The air is returned through registers located around the walls and columns. The duct work in these areas conforms with standard practice.

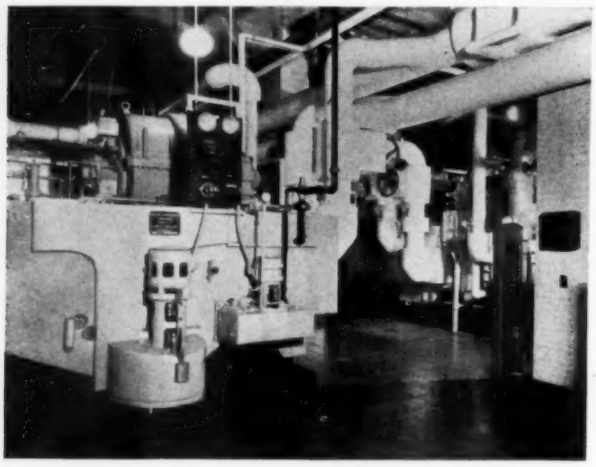


Fig. 7—View of refrigeration plant

The refrigeration equipment (Fig. 7) for the entire air conditioning system consists of four centrifugal machines each driven by a 350 hp variable speed motor. These machines, each weighing 30 tons, are located in the fourth subbasement. Three centrifugal pumps (1800 gpm) are located adjacent to the refrigeration machines. Two of these pumps carry the entire load, the third is an auxiliary. The pumps and intercoolers are shown in Fig. 8. The water to be cooled is forced by these pumps through the coolers of the refrigeration machines into a 12-in. all welded pipe system which carries the water to the fifteenth floor where the piping system branches into four 8-in. lines, each of which, in turn, supplies a booster pump. These booster pumps are located at each of the four dehumidifiers. The water is forced through the sprays of the dehumidifiers by the booster pumps and is carried by gravity through individual lines leading from the collecting pans of the dehumidifiers to an 11,000 gallon surge tank located on the thirteenth floor. The water returns from the surge tank to the suction side of the pumps located in the fourth subbasement through a 12-in. pipe line to complete the refrigerated water circulating circuit.

The water for the system supplying the lunch rooms is chilled by means of three intercoolers located in the refrigeration room and inserted into the cold water supply line leading from the refrigeration machines to the fifteenth floor. Each subbasement dehumidifier is connected with an independent intercooler. The water to be chilled is passed through the intercooler prior to forcing it through the sprays.

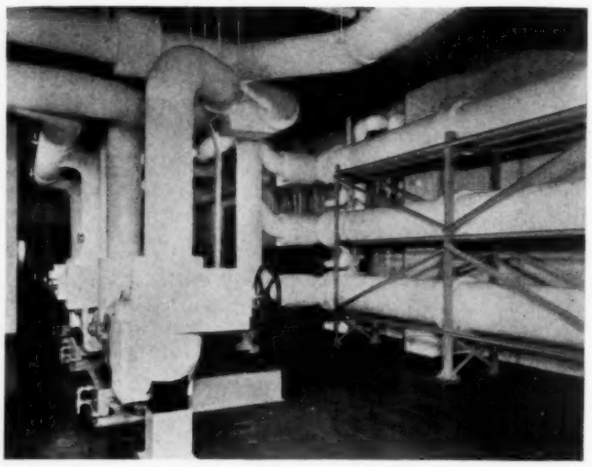


Fig. 8—Pumps and intercoolers

The following is a summary of the magnitude of the system:

Air supplied	540,000 cfm
Square feet floor area conditioned	650,000
Refrigeration	1,350 tons
Cold water circulated	3,400 gpm
People	6,500
Seating capacity restaurants	2,500
Lunches served per day	7,800
Electrical power connected	2,170 hp
Fans	71
Dehumidifiers	7
Pumps	10

John H. Daly

John H. Daly, a Member of the Society since 1915, died suddenly of heart failure on March 18 at the age of 53 at Denver, Colo. He was always very much interested in heating and ventilating, and spent considerable time in their study.

Mr. Daly was born in Massachusetts and in his early years moved to Akron, Ohio, where he lived for a short time. Upon moving to Denver, he entered the plumbing trade and several years later he established his own business as a plumbing and heating contractor. When injury resulting from a fall prevented him from continuing his own business, he became associated with the Morse Brothers Machinery Co. as salesman and later with the Hedges-Atkin-McGee Co., selling plumbing and heating equipment. This company became the Denver branch of the Standard Sanitary Manufacturing Co. Following this connection, he was associated with the Spencer Heater Co. until the government took over the company's plant during the war, necessitating the closing of the Denver branch.

Mr. Daly then became the representative for several eastern manufacturers, working in this capacity for about two years. In 1921 he incorporated the Daly Co., which was distributor throughout the mountain states for nationally known manufacturers of coal and gas fired boilers, gas appliances, heating specialities and ventilating equipment.

The Officers and Council regret the passing of Mr. Daly, who was an active member and helped in the organization of a Denver chapter, and have sent an expression of their sympathy to his sister, Mrs. J. Kane, of Providence, R. I., who survives.

The Auditorium

PROGRAM SEMI-ANNUAL MEETING

1934

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

THE INN — **BUCK HILL FALLS, PA.**

JUNE 19 to 22, 1934

(All Events on Daylight Saving Time)

Tuesday, June 19, 1934

9:30 a. m. Guide Publication Committee Meeting.
1:30 p. m. Meeting of the Committee on Research.
7:00 p. m. Dinner and Meeting of the Council.

Wednesday, June 20, 1934

9:00 a. m. Registration.
9:30 a. m. Greeting: C. V. Haynes, President.
Technical Papers:
Criteria for Industrial Exhaust Systems, by J. J. Bloomfield.
Dry Bulb vs. Effective Temperature Control, by A. E. Beals.
Influence of Stack Effect on the Heat Loss from Tall Buildings, by Prof. Axel Marin.
Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet.
12:30 p. m. Get-together Luncheon (Main Dining Room). Address: Examining the Construction Recovery Program, by Malcolm Muir, Pres., McGraw-Hill Publishing Co., Inc.
1:30 p. m. Men's Golf Tournament—Buck Hill Course.
2:00 p. m. Ladies' Swimming Party at the Pool.
7:00 p. m. Beefsteak Dinner at Skytop with *A. S. R. E.* Members and Ladies.

Thursday, June 21, 1934

8:30 a. m. Breakfast Meeting of Nominating Committee.
Morning: Morning Setups for Ladies—Bowling on Green or Swimming in the Pool.
9:30 a. m. JOINT SESSION OF *A. S. R. E.* and A. S. H. V. E. W. H. Carrier, presiding.
Technical Papers by *A. S. R. E.:*
The Water Vapor Centrifugal Compressor, by Paul Bancel.
Electric Rate Making for Small Air Conditioning Units, by D. W. McLenegan.
10:30 a. m. Intermission.
Address: The Reemployment of the Construction Industry, by Thomas S. Holden, Vice-Pres., F. W. Dodge Co.
Technical Papers by A. S. H. V. E.:
Operating Results of an Air Conditioning System Compared with Design Figures, by J. R. Hertzler.
What Is the Cooling Load Factor in Air Conditioning?, by John Everetts, Jr.
1:30 p. m. Bridge Luncheon for Ladies at Buck Hill Tennis Club.
Research Cup Tournament for A. S. H. V. E. Members at Buck Hill Course.
7:45 p. m. Banquet, Entertainment and Dance (Main Dining Room).

Friday, June 22, 1934

9:30 a. m. Amendments to Constitution and By-Laws.
Technical Papers:
Insulating Value of Bright Metallic Surfaces, by Prof. F. B. Rowley.
Heat Transfer from Direct and Extended Surfaces with Forced Air Circulation, by Prof. G. L. Tuve and C. A. McKeeman.
Factors Affecting the Heat Output of Convectors, by Prof. A. P. Kratz, M. K. Fahnestock and E. L. Broderick.
Morning: Automobile Trip for Ladies of A. S. H. V. E. and *A. S. R. E.* to Points of Interest in the Poconos.
12:30 p. m. Luncheon—(Main Dining Room).
Afternoon: Trip for Ladies to Skytop for Duplicate Bridge and Refreshments.
2:00 p. m. Golf for Men at Skytop Course.

Method of Choosing Location of, Financing and Conducting Meetings of the Society

Resolved: That inasmuch as the Annual and Semi-Annual Meetings of the Society come under the jurisdiction of the Council, the following rules governing the handling of such meetings be adopted by the Council and published in the JOURNAL of the Society at least twice during every year, preferably just prior to each meeting:

1—The Council will select the city in which the Annual or Semi Annual Meeting is to be held, giving due consideration to the invitations received from Chapters or members as well as to the advisability of so distributing those meetings as to make them of the greatest advantage to the general membership, and to reduce as far as possible the expense of members attending.

2—That an appropriation be made to cover the entertainment or local expenses, incurred in connection with the meeting not exceeding $500.00, the regular meeting expense to be taken care of by the General Fund of the Society in the regular way.

3—That no registration fee or compulsory obligations of any nature be imposed on members or guests.

4—That the purchase of tickets for banquets or for any other form of entertainment that may be provided be entirely voluntary.

5—That the grouping of features and the sale of tickets for group features be discouraged.

6—That the raising of Funds from manufacturers of heating apparatus be discouraged.

7—That the display of samples, or of literature, advertising the product of any manufacturer in any way, shape or form, be not permitted at the booths, registration desk, or in or about the meetings.

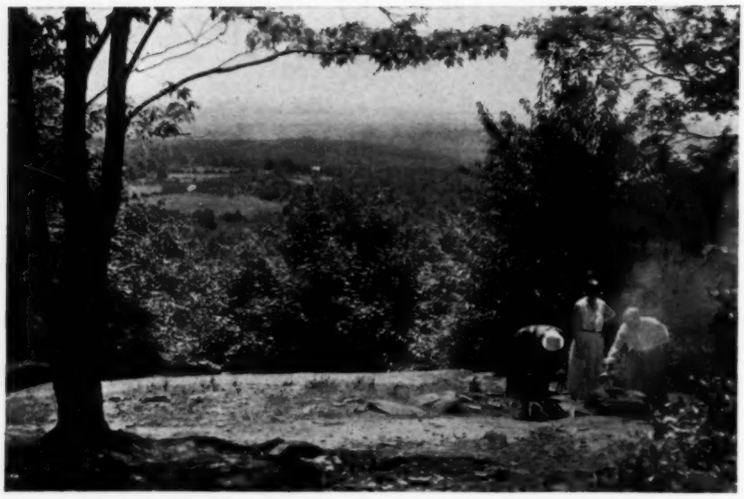
8—That the distribution of trade papers be entirely at the discretion of the committee in charge.

9—That the local Chapter, or local members, be empowered to form a General Committee with such sub-committees as may be required to handle the details of transportation, hotel accommodations, entertainment, finance, etc., and that this General Committee be requested to confer frequently with the Council, through the Secretary of the Society, and to make frequent reports on progress in connection with the various matters being handled by them.

10—That the arrangements of elaborate and costly entertainment features be discouraged.

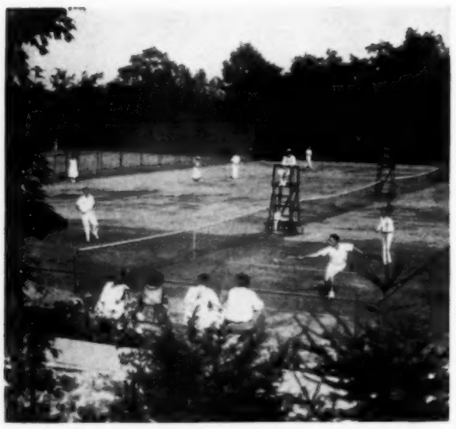
Adopted at Council Meeting, January 29, 1926.

BUCK HILL FALLS



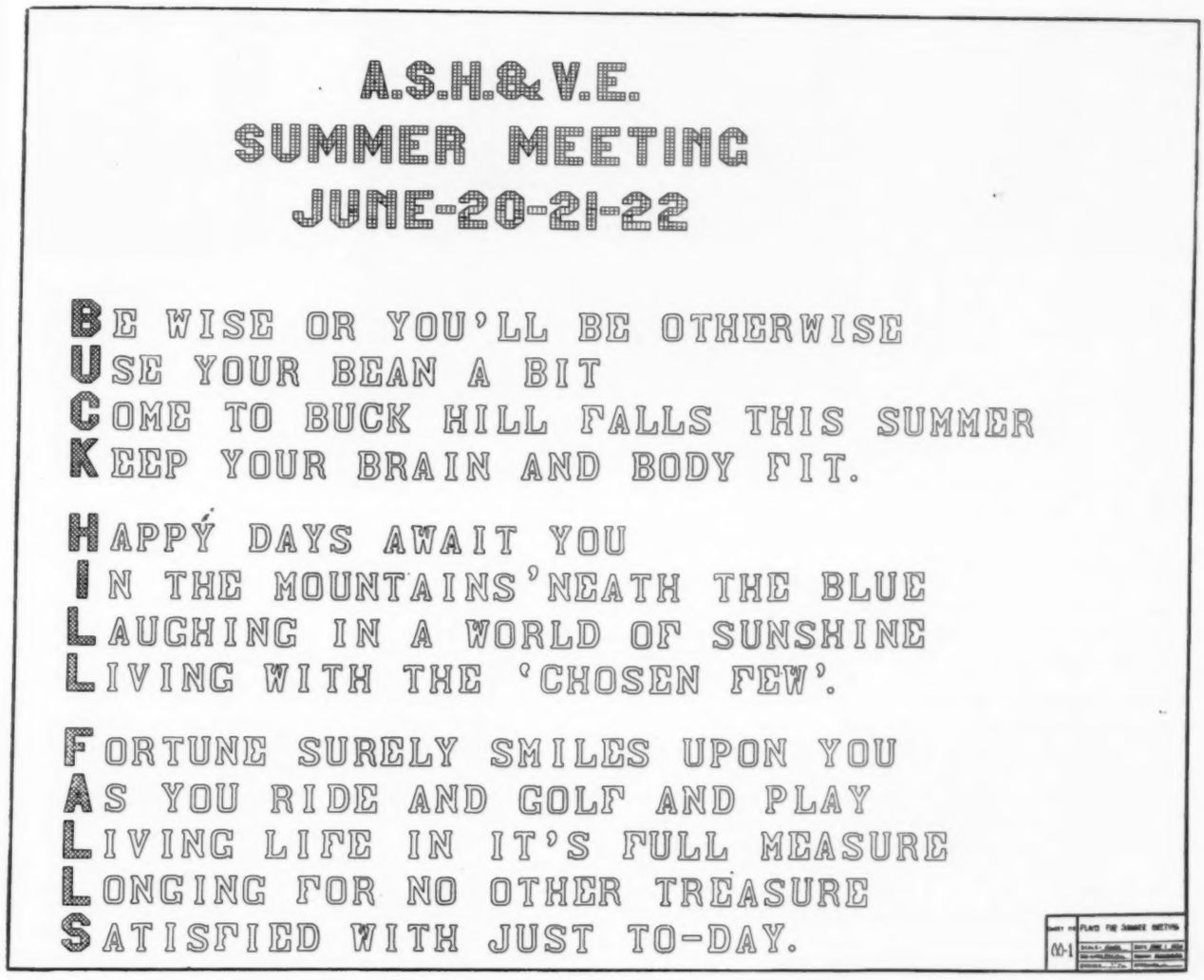
A Picnic on the Mountain

A Morning Ride



Tennis Match

Putting for a Birdie


A.S.H.& V.E.
SUMMER MEETING
JUNE-20-21-22
BE WISE OR YOU'LL BE OTHERWISE
USE YOUR BEAN A BIT
COME TO BUCK HILL FALLS THIS SUMMER
KEEP YOUR BRAIN AND BODY FIT.
HAPPY DAYS AWAIT YOU
IN THE MOUNTAINS 'NEATH THE BLUE
LAUGHING IN A WORLD OF SUNSHINE
LIVING WITH THE 'CHOSEN FEW'.
FORTUNE SURELY SMILES UPON YOU
AS YOU RIDE AND GOLF AND PLAY
LIVING LIFE IN IT'S FULL MEASURE
LONGING FOR NO OTHER TREASURE
SATISFIED WITH JUST TO-DAY.
00-1

Plans and Specifications* Covering The Building of a Successful Summer Meeting for the Members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

1. It is the intent of these specifications to create further interest and participation in the construction of one successful semi-annual meeting on the site herein described.

2. All previous communications on subject addressed your chapter, as well as all data sent members individually are to be considered a part of these specifications.

3. *Location of Site*—In Pocono Mountains, at Buck Hill Falls, Pa., one hundred miles north of Philadelphia.

4. *Description of Plot*—Area: three thousand acres. Mountainous country, heavily timbered, mountain streams, golf courses, etc.

5. *Existing Facilities*—The Inn, of ample size for requirements, golf course, tennis courts, stables, bowling greens, etc., all of which are to be made available by the owner for certain fixed consideration elsewhere described.

6. *Completion Date*—Actual construction of the meeting is to start the morning of June 20th and is to be officially completed at noon of June 22nd. All participants may at their option, arrive before and depart after these dates without added penalties in increased rates for space occupied.

7. *General Procedure*—Each member should present himself and family at site not later than morning of the 20th for welcome by members of the engineer's staff.

8. *Field Office*—Suitable space (size and type arranged by reservation) will be assigned him by the owner. Each space so assigned to be provided with electric light, water and sewer connections.

9. *Tools and Equipment*—If, in the opinion of the member, such tools as golf clubs, tennis racquets, fishing tackle, spirituous liquors, etc., are required to comply with the intent of this specification, they are to be furnished by him.

10. *Access to Site*—All rail shipments to be D. L. & W. Delivery at Cresco, Pa. For best results automobile traffic will follow road information shown on maps furnished by engineer's office.

11. *Extras*—All work to be completed at advertised rates. No additional charges covering steak fry, beer party and banquet will be allowed.

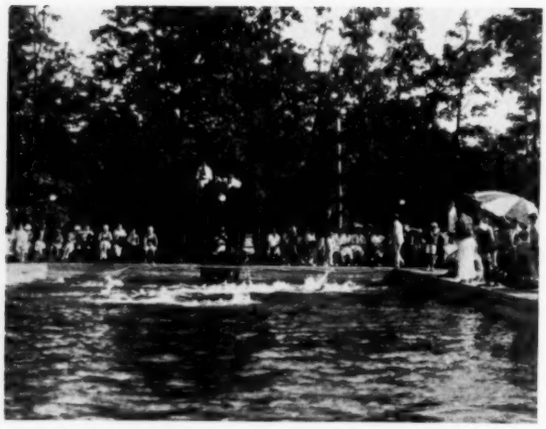
12. *Cleaning Up*—All divots to be replaced and premises in general to be left in substantially the same shape as found at start of construction.

13. *N. R. A. Compliance*—All codes applying whether signed, written or thought of will be strictly observed—but don't let that keep you away.

14. *Penalties*—All A. S. H. & V. E. members not participating in this undertaking will be penalized one opportunity of seeing their many friends who will be there, one opportunity of showing their interest in their society and one opportunity of enjoying a delightful vacation at an extremely reasonable cost.

*Engineers, Philadelphia Chapter A. S. H. & V. E., 4210 Sansom St., Philadelphia, Pa.

R. C. Bolsinger
General Chairman of Arrangements



The Buck Hill Pool

W. P. Culbert
Chairman, Golf Tournament Committee

WHEN members and guests of the Society meet at The Inn at Buck Falls, Pa., during June 19 to 22, they will find that all arrangements have been made to make their stay pleasant and enjoyable.

Located high in the heart of the Pocono Mountains, Buck Hill Falls is noted for its scenic beauty and offers a natural layout of mountain valleys, waterfalls, small lakes, and winding trails. It is easily accessible by rail, motor or air and detailed information on transportation is given on page 283. In the Poconos summer temperatures and humidities are low so that visitors are able to enjoy the outdoor life and sports that are available.

The Inn offers comfortable rooms and pleasant surroundings at moderate rates and A. S. H. V. E. members have received a reservation card which should be returned promptly, so that they may be shown to their accommodations immediately upon arrival. Guests may indulge in swimming, fishing, and lawn bowling, for which there is no charge, and in golf, riding, and tennis at nominal rates.

A fine program has been planned and several events will be held jointly with members of the *A. S. R. E.* who are meeting at the same time at Skytop Lodge, which is nearby. On Tuesday, meetings of the Guide Publication Committee, the Committee on Research, and the Council will be held in advance of registration and the opening session on Wednesday morning.

At the first session papers will be presented on Criteria for Industrial Exhaust Systems by J. J. Bloomfield, Dry Bulb vs. Effective Temperature Control, by A. E. Beals, Influence of Stack Effect on the Heat Loss from Tall Buildings, by Prof. Axel Marin, and Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet. An address by Malcolm Muir, president of McGraw-Hill Publishing Co. and former N. R. A. deputy administrator, on Examining the Construction Recovery Program will be an innovation at the Get-Together Luncheon on Wednesday at noon. While the men are enjoying a golf tournament on the Buck Hill Course, the ladies will have an afternoon swimming party at the pool. On Wednesday evening at 7 o'clock the A. S. H. V. E. group will join the members and ladies of the *A. S. R. E.* for a beefsteak dinner at Skytop.

Early morning activities are planned for Thursday when members of the Nominating Committee will meet at breakfast and ladies can enjoy morning setups, bowling on the green, or swimming in the pool. At 9:30 a joint session of the A. S. H. V. E. and *A. S. R. E.* will be held with W. H. Carrier, past president of both Societies, presiding. The complete program of technical papers appears on page 279.

In the afternoon ladies will have a bridge luncheon at the Buck Hill Tennis Club. At the same time the Research Cup Tournament for A. S. H. V. E. members will be played at the Buck Hill Course. Thursday evening is the date of the Semi-Annual Banquet, which will include entertainment and dancing.

The Friday morning session at 9:30 will be devoted to research reports.

During the morning an automobile trip for ladies of the A. S. H. V. E. and the *A. S. R. E.* will take them to points of interest and beauty in the Poconos. At 12:30 luncheon will be served and following this ladies will go to Skytop for an afternoon of duplicate bridge and men will have an opportunity to test their skill on the Skytop golf course.

The reduced railroad fares and the moderate hotel rates will enable members and their families to have a pleasant vacation at low cost. The Committee on Arrangements of the Philadelphia Chapter has worked hard to provide a program of interest and entertainment and they are looking forward to welcoming A. S. H. V. E. members to Buck Hill Falls and acting as their hosts from June 19-22.

Hotel Rates at The Inn

	Per Person Per Day
Double room with bath—twin beds—two persons	$6.50 and $7.50
Double room with lavatory and toilet—twin beds—two persons	6.00
Double room with lavatory—twin beds—two persons	5.00
Single room with bath—one person	7.50
Single room with lavatory and toilet—one person	6.50
Single room with lavatory—one person	5.00
Children (under 10) in room with parents	3.00

All rates American Plan—include meals.

M. F. Blankin
Banquet

W. R. Eichberg
Entertainment

J. H. Hucker
Transportation & Publicity

W. F. Smith
Finance

Transportation

Travel by Train

Identification certificates have been issued to members of the Society from the Secretary's office which will permit them to purchase reduced round trip tickets from all points in the U. S. and Canada at a rate of one fare and one-third.

The Identification Certificate *must be presented to ticket agents within the period specified for the different territories* and the return trip must be made within 30 days in addition to date of sale:

Central Passenger & Trunk Line Assn.: East of and including Chicago and St. Louis north of the Ohio and Potomac Rivers to the Atlantic Seaboard, south of New England and Canada. (June 15-21.)

New England Passenger Assn.: From New England States. (June 15-21.)

Southeastern Passenger Assn.: From territory south of Ohio and Potomac Rivers and east of Mississippi River. (June 14-20.)

Western Passenger Assn.: From territory west of Chicago, Peoria, and St. Louis and east of the States of Washington, Oregon and Nevada. (June 13-21.)

Southwestern Passenger Assn.: From territory southwest of St. Louis, including Texas, Arkansas, Oklahoma, Missouri (south of Missouri River), and Louisiana (west of Mississippi River). (June 13-19.)

Trans-Continental Passenger Assn.: From California, Nevada, Oregon and Washington. (June 11-19.)

Canadian Passenger Assn. (Eastern Lines): From territory east of and including Armstrong, Fort William and Sault Ste. Marie, Ont. (June 15-21.)

Canadian Passenger Assn. (Western Lines): From territory west of Armstrong, Fort William and Sault Ste. Marie, Ont. (June 14-20.)

Round trip tickets will be issued for going and returning same route, unless diverse routes are specified at the time ticket is purchased. The return portion of the ticket *must be validated at the regular railroad ticket office at Cresco, Pa.,* at the time of starting return trip.

Children of five years and under twelve years of age will be charged one-half of the adult fare.

Only one Identification Certificate is needed for each member of the Society, as the certificate also takes care of members of the family.

The list of fares quoted are the lowest available to Cresco, Pa.:

From	One way fare direct	Convention fares both ways direct	Pullman lower berth
New York, N. Y.	$ 3.45	$ 4.60	....
Buffalo, N. Y.	10.85	14.47	$ 3.00
Toronto, Ont.	14.52	19.36	3.75
Philadelphia, Pa.	4.13	5.51	....
Pittsburgh, Pa.	....	....	6.75
Via Buffalo	18.30	24.40	....
Via Manunka Chunk	16.24	21.66	....
Washington, D. C.	9.03	12.04	....
Cincinnati, Ohio	23.94	31.92	6.00
Cleveland, Ohio	17.48	23.31	4.50
St. Louis, Mo.	33.55*	44.74*	10.13
Detroit, Mich.	19.85	26.47	6.00
Grand Rapids, Mich.	24.76	33.02	7.50
Chicago	28.16*	37.55*	8.25
Milwaukee	30.71*	40.95*	9.00
Minneapolis	40.37*	53.83*	10.75
St. Paul	40.05*	53.40*	10.75
Kansas City	41.95*	55.94*	12.00
Indianapolis, Ind.	26.13*	34.84*	8.25
Dallas			15.63
Via St. Louis	46.88*	62.51*	....
Via Washington	51.05*	68.07*	....
New Orleans			
Via Washington	42.53*	56.71*	....
Atlanta, Ga.	28.18*	37.58*	....
Louisville, Ky.	28.25*	37.67*	....

	One Way	45-Day	Season	Pullman L. B.
†Seattle	$92.51*	$122.70	$132.15	$24.75
†San Francisco and Los Angeles, Calif.	93.34*	122.70	132.15	24.75

*A slightly higher rate is in effect if the trip is made via New York.

†S. T. fares to Cresco are based on $2.80 (which is double one-way fare) from Scranton, Pa.

Motorists

As motorists approach Cresco on main roads from various cities, they will notice blue road signs marked *Buck Hill Falls,* which will direct them to The Inn. The accompanying map shows route numbers and gives direct routes from New York and Philadelphia:

New York City to The Inn at Buck Hill Falls

Via Holland Tunnel (distance from N. J. end of tunnel 100.3 miles). Follow U. S. and State route numbers: N. J. No. 9 from Jersey end Holland Tunnel to mileage 18.00—N. J. No. 6 to State Line at the Delaware River—U. S. 611 to D. L. & W. station in East Stroudsburg, Pa., and junction with U. S. 209. Follow U. S. 209 1.5 miles and at junction bear sharp left onto Pa. 190, which in 2.1 miles becomes Pa. 90. At Mountainhome—mileage 98.8—bear sharp right onto 390 and continue for 1.0 mile, taking left fork and leaving Pa. 390. Continue .5 miles to The Inn.

Via George Washington Bridge (distance from N. J. end of bridge 104.3 miles). A good optional route from the Bronx and upper Manhattan: At N. J. end of the bridge follow N. J. 4 to Paterson. At junction with N. J. 6 bear right and continue to State line at Delaware River. Follow directions above for route through East Stroudsburg to Buck Hill Falls.

Philadelphia to The Inn at Buck Hill Falls

Via Delaware Water Gap (distance from City Hall 102.4 miles). Follow U. S. and State route numbers: From City Hall follow U. S. 611 to Willow Grove, Doylestown, Easton and Martin's Creek. At center of Martin's Creek turn sharp right cross bridge and continue on Pa. 827 to junction with U. S. 611. Follow latter route through Portland, Water Gap to East Stroudsburg and junction with U. S. 209. Follow 209 for 1.5 miles and at junction with Pa. 190 bear sharp left. The route in 2.1 miles becomes Pa. 90. At Mountainhome bear sharp right onto Pa. 390 and continue for 1.0 mile, taking left fork and leaving Pa. 390. Continue 0.5 mile to The Inn.

Via Easton and Wind Gap (distance from City Hall 97.1 miles): From City Hall follow U. S. 611 to Willow Grove, Doylestown and Easton. Three blocks beyond Square keep left and ascend College Hill on Pa. 102. At Stockertown continue straight through, taking Pa. 12 to Snydersville. Turn sharp left at Snydersville and continue for 3.0 miles to Bartonsville. Join U. S. 611 and follow to Swiftwater. Turn sharp right, keep left at fork .2 mile and take left turn 1.1 miles further. At junction with Pa. 615 turn sharp right and left .2 mile. Continue to Mountainhome on Pa. 90, bearing right onto Pa. 390. Follow to fork 1.0 mile, where bear left and arrive at The Inn in .5 mile.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 9 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by June 15, 1934, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

Candidates	*Proposers*	*Seconders*
BORLING, J. R., Engr.-Custodian, Board of Education, Chicago, Ill.	John Howatt J. J. Finan	J. H. O'Brien J. F. Tobin
FITZSIMONS, J. P., Engr. and Estimator, Robt. Fitzsimons Co., Ltd., Hamilton, Ont., Canada. (*Advancement.*)	W. A. Duncan W. B. Pennock	G. E. Cole J. W. O'Neill
GAULT, George W., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys T. F. Rockwell	C. E. Wright E. W. Cheeseman
HAYTER, BRUCE, Chief Engr., Inst. of Thermal Research, American Radiator Co., Yonkers, N. Y.	W. R. Zuhlke W. W. Timmis	R. E. Daly C. W. Brabbée
HOLMES, R. E., Air Cond. Engr., Westinghouse Elec. & Mfg. Co., Springfield, Mass.	C. A. M. Weber (*Non-Member*) R. E. Cross	J. H. Ashbaugh (*Non-Member*) M. Kalischer (*Non-Member*)
MORSE, FLOYD W., Asst. Gen. Sales Mgr., Chamberlin Metal Weather Strip Co., New York, N. Y.	H. L. Hosking W. W. Timmis	J. K. Peacock E. C. Shepard
POWERS, EDGAR C., Dist. Mgr., Chase Brass & Copper Co., Htg. Prod. Div., Philadelphia, Pa.	A. C. Caldwell H. G. Black	C. B. Eastman M. F. Blankin
REED, I. G., Asst. Supt. and Chief Engr., Grant Bldg., Inc., Pittsburgh, Pa.	V. W. Hunter J. L. Blackshaw	P. A. Edwards H. G. Bucher
STEGGALL, HOWARD B., Br. Mgr., U. S. Radiator Corp., Pittsburgh, Pa.	H. A. Beighel H. B. Orr	F. C. McIntosh M. A. Benson

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

ADAMS, EUGENE I., Chief Engr., Michigan State College, Lansing, Mich.

BENOIST, LEROY L., Mgr. and Htg. Engr., Benoist Bros., Mt. Vernon, Ill.

FERGUSON, R. R., Mgr., Trade Dept., American Blower Corp., New York, N. Y. (*Reinstatement and Advancement.*)

HEBLEY, HENRY F. J., Research Engr. and Member of Firm, Allen & Garcia Co., Cons. Engrs., Chicago, Ill.

HODGE, WILLIAM B., Vice-Pres., Parks-Cramer Co., Charlotte, N. C.

MAILLARD, ALBERT L., Head of Air Cond. Div., Kansas City Power & Light Co., Kansas City, Mo.

MERRILL, FRANK A., Htg. & Vtg. Engr., Office of Hollis French, Cons. Engrs., Boston, Mass.

MILES, JAMES C., Dept. Mgr., The Henry Furnace & Foundry Co., Cleveland, Ohio. (*Reinstatement.*)

THOMAS, L. G. LEE, Vice-Pres., Economy Pumping Machinery Co., Chicago, Ill.

WALKER, EDMUND R., Htg. Div., Fedders Mfg. Co., Inc., Buffalo, N. Y.

WASHINGTON, GEORGE, Engr., Hoffman Specialty Co., Inc., Chicago, Ill.

WOHL, MAURICE W., Engr., American Insulating Corp., Brooklyn, N. Y.

WOODS, E. H., Owner, F. H. Higgins, Ithaca, N. Y.

ASSOCIATES

ANDES, WILLIAM, Secy.-Treas., The Andico Co., Cleveland, O.

ELLIOTT, N. B., Br. Mgr., American Blower Corp., Milwaukee, Wis.

GAY, L. M., Power Engr., Texas Power & Light Co., Dallas, Texas.

GOLDBERG, MOSES, Pres., Electric Motors Corp., New York, N. Y.

HUNZIKER, CHESTER E., Br. Mgr., American Blower Corp., Schenectady, N. Y.

MATTHEWS, J. E., Dist. Mgr., B. F. Sturtevant Co., Kansas City, Mo.

POTVIN, LEO J., Dist. Office Mgr., Hoffman Specialty Co., Inc., Chicago, Ill.

RODGERS, FREDERICK A., Br. Mgr., Minneapolis-Honeywell Regulator Co., Cleveland, O.

JUNIORS

GILES, ALFRED F., Vent. Engr., H. H. Robertson Co., Pittsburgh, Pa.

HYDE, L. L., Gen. Mgr., M. J. O'Neil, St. Paul, Minn.

MORRIS, ARNOLD M., Industrial Dept., U. S. Navy Yard, Philadelphia, Pa.

RAYMER, WILLIAM F., Jr., Sales Engr., American Blower Corp., Newark, N. J.

STUDENTS

BENSE, WILLIAM M., Student, Pratt Inst., Brooklyn, N. Y.

DEAN, FRANK J., JR., Graduate Student, Mass. Inst. of Tech., Cambridge, Mass.

KAMPISH, N. S., Student, Pratt Inst., Brooklyn, N. Y.

KEPLER, DONALD A., Student, Pratt Inst., Brooklyn, N. Y.

KEYS, L. F., Student, Pratt Inst., Brooklyn, N. Y.

PRIESTER, GAYLE B., Student, Harvard Graduate School, Boston, Mass.